ASHRAE OFFICIAL PUBLICATION

JOURNAL

Heating ✦ Refrigerating ✦ Air Conditioning ✦ Ventilating

AMERICAN SOCIETY OF HEATING, REFRIGERATING AND AIR-CONDITIONING ENGINEERS

MAY 1961

MAY
1961

VOL. 3 NO. 5

JOURNAL

OFFICIAL PUBLICATION

Formerly Refrigerating Engineering including Air Conditioning, and incorporating the ASHAE Journal.

FEATURE

NEWS

DEPARTMENTS

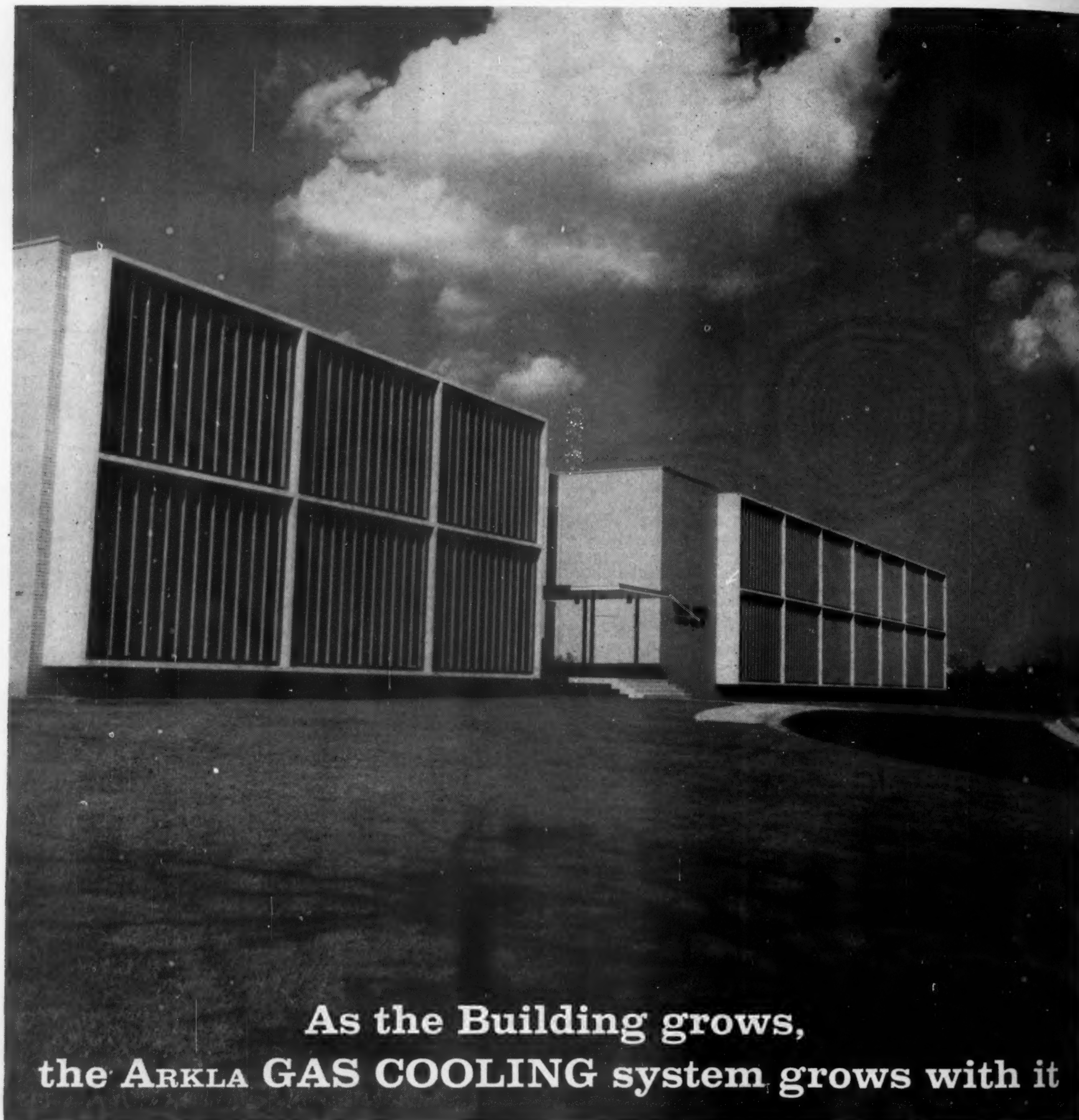
Member
Audit Bureau of Circulations

Published by the American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc.
Editorial and Circulation Offices, 234 Fifth Avenue, New York 1, N. Y. MUrray Hill 3-6496
Advertising Offices, 62 Worth Street, New York 13, N. Y. BArclay 7-6262
Midwest Advertising Office, 35 East Wacker Drive, Chicago 1, Ill. Financial 6-7255
Southern Advertising Representative, James C. Crawford, 921 Fulton Federal Building, Atlanta, Ga. JAckson 2-9769
Western Advertising Representative, Chris Dunkle & Associates, 420 Market Street, San Francisco 11, Calif. SUtter 1-8854
740 South Western Avenue, Los Angeles 5, Calif. DUnkirk 7-6149

Price by subscription, $5.00 1 yr.; $8.00 2 yr.; $10.00 3 yr. in U.S.A. and Canada, $2.00 per yr. additional in other countries (payable with order). Single copies (payable with order) 75c; 6 months old or more $1.00; 5 yr. old or more $1.50. Published monthly.
Volume 3 includes the January through December 1961 numbers.

Second Class Postage paid at East Stroudsburg, Pa. Postmaster: On undeliverable copies send Form 3579 to ASHRAE JOURNAL, 234 Fifth Avenue, New York 1, N. Y.

Hughes Printing Company, East Stroudsburg, Pa.



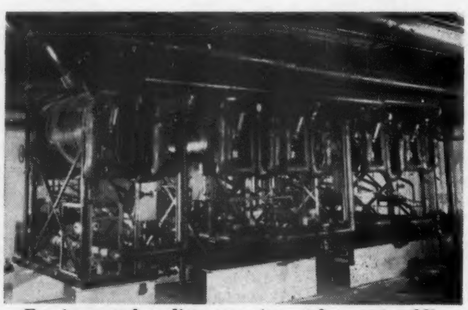
Architect: Folger & Pearson; Mechanical Contractors: Troug & Nichols. Modern gas cools and heats this headquarters building of Yellow Transit Freight Lines in Kansas City, Missouri. The Arkla Gas air conditioning unit uses the same gas-fired boiler that heats in winter to cool in summer.

The headquarters building of Yellow Transit Freight Lines, Kansas City, Missouri, was designed to take a third story without major alterations. That's why they chose Arkla gas cooling units . . . a system that can "grow" quickly and at low cost.

When the building expands, they'll just add an Arkla unit. They go on the line right next to the rest, using the same basic piping – and steam from the same gas-fired boiler that energizes all the Arkla units.

The present cooling system includes five 25-ton Arkla Gas Absorption Water Chillers. These versatile units use steam from the gas-fired boiler to provide chilled water for comfort cooling. The same boiler heats in winter. And thrifty gas keeps fuel costs low.

For specific information on Arkla gas air conditioning, call your local Gas Company. Or write Arkla Air Conditioning Corporation, General Sales Office, 812 Main St., Little Rock, Arkansas. *American Gas Association.*

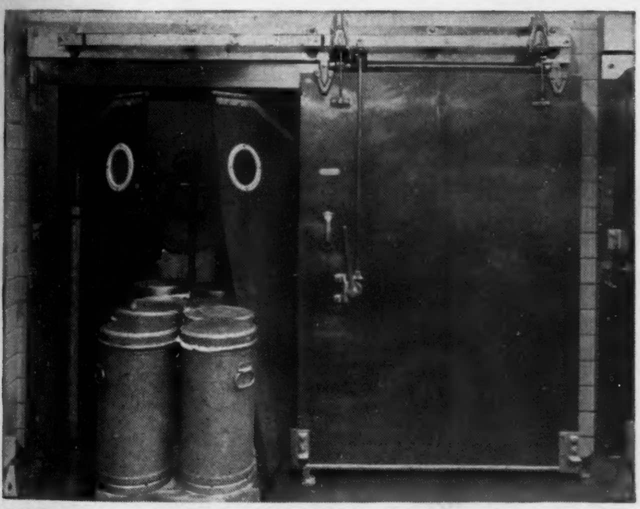


For increased cooling capacity, at low cost, additional Arkla units can be installed.

FOR HEATING & COOLING... GAS IS GOOD BUSINESS!

FAST OPENING AND CLOSING. Jamison Electroglide® Power Doors speed traffic to loading dock—single leaf cooler door, left, and bi-parting freezer door, right.

New $3,000,000 automated plant features Jamison Horizontal Sliding Doors



SAVES REFRIGERATION. Jamison Manual Horizontal Sliding Door in combination with self-closing Flexidor®.

RAPID, CONVENIENT LOADING. Jamison Manual Horizontal Sliding doors facilitate loading of dairy products.

• In a dramatic advance toward automated processing of fluid milk and ice cream products, the Borden Company recently completed a new $3,000,000 plant at Milwaukee, Wisconsin. An important feature of this modern plant is a number of Jamison Horizontal Sliding Cold Storage Doors.

More and more, in the design of new, modern facilities like this one, Jamison Doors are specified to save space, minimize refrigeration loss, speed traffic and facilitate handling of products and materials. Get the story of the cost-saving benefits of Jamison Doors from your architect, or write to Jamison Cold Storage Door Co., Hagerstown, Md.

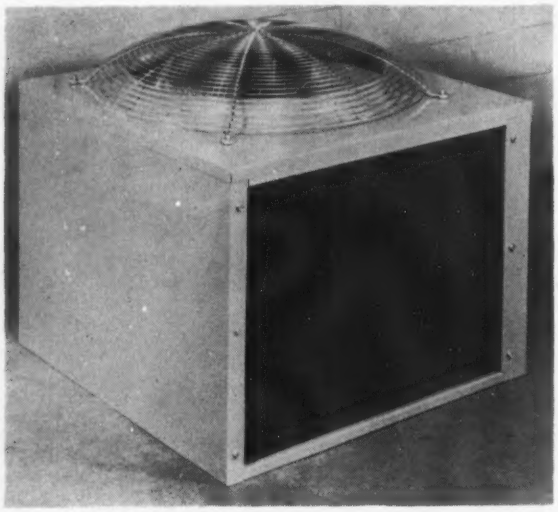
JAMISON

COLD STORAGE DOORS

PARTS AND PRODUCTS

AIR COOLED CONDENSING UNIT

Especially designed for small and medium sized homes and for use in multi-roomed apartments, Model 1202-03 has a capacity of 22,000 Btu/hr for split



system installation. An evaporator fan relay allows continuous operation of the indoor fan for circulation or fan operation only when the compressor is running. Installation is facilitated by a sub-cooling circuit, since refrigeration lines no longer require soldering.

Chrysler Corporation, Airtemp Div, P. O. Box 1037, Dayton 1, Ohio.

EXPANSION COMPENSATORS

Screwed, welded or flanged ends; heavy duty two-ply bellows; positive anti-torque mechanism; and protective shroud of Model HS Expansion Compensators are all of stainless steel. This permits use under corrosive conditions, at working pressures to 175 psi and temperatures to 750 F. Unit is manufactured in pipe sizes ¾ to 3 in.

Also available is an all-bronze compensator, primarily for applications where the use of dissimilar metals might cause corrosion problems. Model HB is designed for working pressures to 150 psi and temperatures to 406 F, and is manufactured in male thread and sweat ends sizes ¾ to 3 in.

Flexonics Corporation, 1315 S. Third Ave., Maywood, Illinois.

INSULATING FILL

Recently developed for cryogenic applications, this insulating fill offers thermal conductivity values lower than materials currently in use. Micro-Cel T-4, a new type of synthetic calcium silicate, has an apparent mean conductivity of 0.100 Btu in./hr sq ft F, as determined between liquid nitrogen and ambient temperature at atmospheric pressure. As an evacuated powder, Micro-Cel grades have apparent mean thermal conductivity values ranging from 5.5 to 7.5 microwatt/cm K as determined at an absolute pressure at one micron Hg.

Completely synthetic, the fill is a free-flowing powder, free of explosion or fire hazard and compatible with commonly used liquid gases, such as oxygen and hydrogen. Combination of low k factor, low density, good handling characteristics and resistance to settling suit it for both stationary cryogenic applications, such as cold boxes, and transport applications, such as refrigerated trucks or trailers.

Johns-Manville Corporation, 22 E. 40th St., New York 16, N. Y.

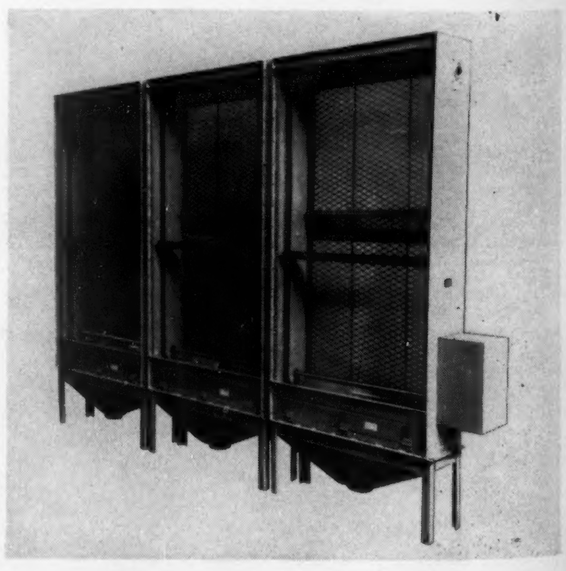
ROOF-TOP COOLING, YEAR-ROUND UNITS

Offered by this manufacturer are a self-contained roof-top cooling unit and combined heating and cooling equipment for offices, stores and other single level commercial buildings. The cooling system is available as a separate unit; the year-round temperature control system is a three-part package consisting of an air-cooled condenser, cooling section and a gas-fired unit for heating. Models are available in 7½ and 10-ton sizes. But one opening through the roof for a combination supply and return air duct is required for the roof-top system.

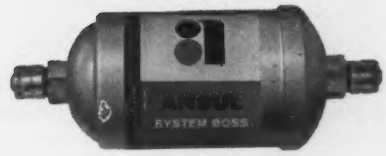
Curtis Manufacturing Company, Refrigeration Div, 1905 Kienlen Ave., St. Louis 33, Mo.

FILTERING, COLLECTION UNIT

For removing heavy concentrations of lint and fibrous material from air in high volume applications, this self-cleaning lint arrester is available in a wide range of sizes to meet air flow requirements from 7000 to 150,000 cfm. Extremely low pressure drop and high arresting efficiency with velocities of 250 to 1900 fpm are cited as features of the Lint-A-Maze. Pressure drop is dependent on which of three standard ar-



rester media is used. Collected material is removed continuously from the face of the filter and deposited

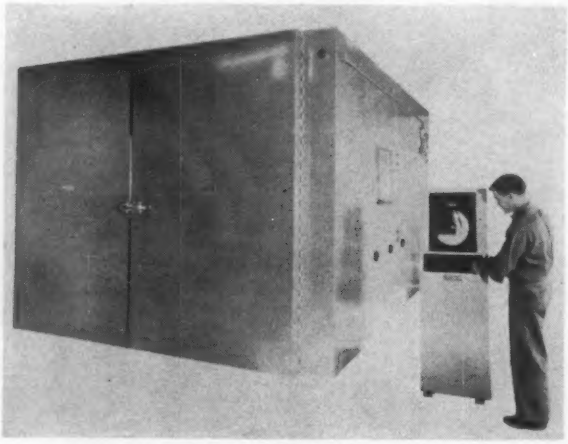


ANSUL
SYSTEM BOSS

in a hopper by a powered sweeper brush. Three standard hoppers are available, for automatic or manual collections.
Rockwell-Standard Corporation, Air-Maze Div, 25000 Miles Rd., Cleveland 28, Ohio.

WALK-IN CHAMBER

Model WF-2100-125+300H, a walk-in environmental chamber with a capacity of 2100 cu ft, provides a temperature range from –125 to 300 F and a relative humidity range from 20 to 95%. Inside unobstructed



usable space is 15 x 20 x 7 ft. Featuring a cascade arrangement and automatic defrost, the chamber has a thermal capacity of 56,000 Btu/hr. There is less than one-F air stratification within the interior. Refrigeration units may be installed in any location within 50 ft from the cabinet. Programmed instrumentation is mounted on a separate console.
Webber Manufacturing Company, Inc., P. O. Box 217, Indianapolis, Ind.

HOT WATER BASEBOARD

Cited as providing even heat throughout rooms, replacing heat where it is lost, eliminating drafts and drying-out blasts and minimizing blower noise and duct rattle, this hot water baseboard is but 7 in. high and 1¾ in. wide. Metal of the unit can be cut with a standard hacksaw, back panels nail to studs and accessories are snapped into place, reducing installation time.
Patco Manufacturing Company, Inc., 231 N. Bread St., Philadelphia 6, Pa.

THREE-HP HEAT PUMP

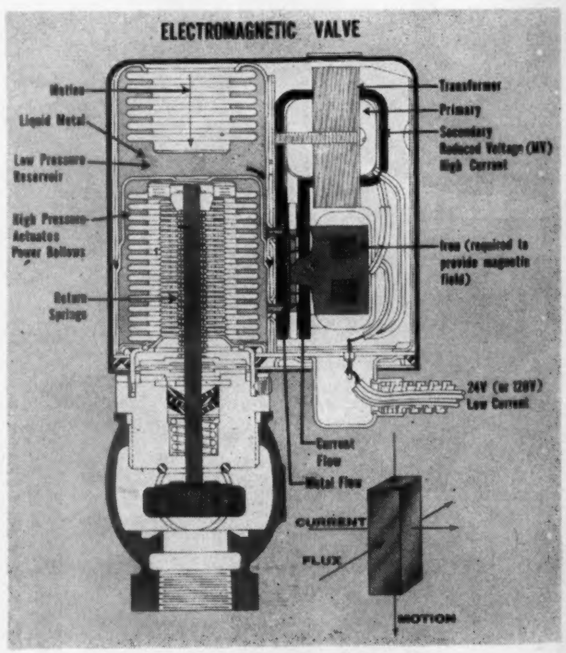
Recently introduced, this three-hp unit is a split system model with a 33,000-Btu/hr cooling capacity and a heating capacity of 32,000 Btu/hr. It is designed for residential and medium sized commercial applications. Featured is a checklight, a bulb which lights on the indoor thermostat if the unit's compressor is not operating.

Supplementary heat is available for Model 3204 in 4.8-kw steps up to 14.4 kw. Outdoor thermostats turn on supplementary heat during the heat pump's defrost cycle to prevent cold air from blowing into the air conditioned space. Indoor blower of the system may be operated independently for air circulation. During the summer the unit will operate at 120 F outside temperature and in winter at –10 F.

A double solenoid reversing valve is located in an insulated compartment to prevent heat loss during the heating cycle. Other features include special expansion valves for both heating and cooling cycles, an insulated refrigerant line and a subcooling pass which allows the indoor unit to be installed at any height above the outdoor unit. A measuring device automatically begins the defrost cycle and a snap control attached to the liquid line terminates the cycle when the coil is defrosted completely. Matched with the 3204 is a Model 1454 evaporator coil unit, which can be installed vertically for an upflow application or horizontally when used with duct work and horizontal free air delivery.
Chrysler Corporation, Airtemp Div, P. O. Box 1037, Dayton 1, Ohio.

ELECTRIC CONTROL VALVE

Using an electromagnetic pump to move liquid metal to control fan-coil units, convectors and radiators, this Fluid-Power valve has but three moving parts: valve stem, power bellows and the liquid metal. Components of the actuator include a field coil, transformer, electromagnetic pump, reservoir bellows and power bellows. High current, up to 180 amp, is supplied by the transformer at a reduced voltage of 30 or 40 millivolt to two electrodes. A magnetic field



between two pole pieces is supplied by the field coil and the power bellows is linked to a conventional valve stem which seals against a disc in the cast body.

The electromagnetic pump is actually a small slit between the electrodes and pole pieces, connecting passages leading to the reservoir and power bellows. Current flows through the field winding and transformer primary, inducing both a current and

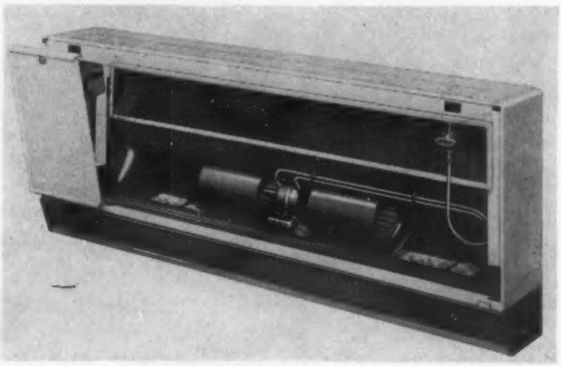
a flux in the liquid metal in the slit. The resultant force pushes the liquid metal through the slit to the power bellows at pressures to 50 psi. Valve stem is forced down by the power bellows, closing the valve. When the current is turned off, a return spring pushes the valve stem and power bellows back to their original positions and the liquid metal flows back into the reservoir bellows.

Unit (shown here before being sealed in epoxy potting compound) will be available in both two and three-way models. The two-way valve can be used with hot water, cold water or steam; the three-way is designed for hot or cold water.

Minneapolis-Honeywell Regulator Company, Commercial Div, 2753 Fourth Ave. S., Minneapolis, Minn.

FAN COIL UNITS

Providing central system air conditioning with individual room controls, Airditioner fan coil units are



available in eleven different types and six colors. Equipped with a high capacity coil and quiet blower fans, the units will heat, cool and dehumidify, ventilate or circulate air in a room. The same piping system, connected to a central source, supplies hot water to the coil for heating and chilled water for cooling. If neither heating nor cooling is required, air can be circulated merely by operating the blower fans. A fresh air intake, with aluminum or steel wall box, is available also.

Air flow capacities are as high as 1500 cfm; an extensive choice of models is available for mounting on the floor, on the wall or concealed in the wall. Filter is located on the front of the unit and pulls out like a drawer. Coils, condensate drain pans and electrical connections are reversible for added convenience in installation. Shown is the interior of the unit.

Modine Manufacturing Company, 1500 DeKoven Ave., Racine, Wisc.

EXPANSION TANK

Installed in the main on the supply side of a boiler, this new model DeAirator-Expansion Tank serves as the only air-eliminating device required on series-loop baseboard systems, as well as an expansion tank and tempering device on circulator start. All heated boiler water passes through it on the way to the heating units. Entrained air and boiler vapors are released by a system of baffles and trapped in the tank. By feeding make-up water into the tank, entrained air is trapped immediately.

General Automatic Products Corporation, 2300 Sinclair Lane, Baltimore 13, Md.

POWER PLANTS

Engineered expressly for air conditioning applications and extensively field-tested, these engines are able to operate for long periods with but routine attention and include both L-head and overhead valve types, for use on all fuels, including Diesel oil and natural gas.

Basic features incorporated are normalized alloy crankcase cast integral with the cylinder block and heavily reinforced and ribbed to prevent distortion; drop-forged and heat-treated alloy steel crankshaft, statically and dynamically balanced, with induction-hardened journals; individual porting; and Stellite-faced intake and exhaust valves with individual rotators. Cooling on all but two series is by standard water-circulating pump, fan belt driven. On the other series, vapor phase cooling is cited as eliminating the pump, assuring fast warmup and maintaining the correct operating temperature.

Continental Motors Corporation, Muskegon, Mich.

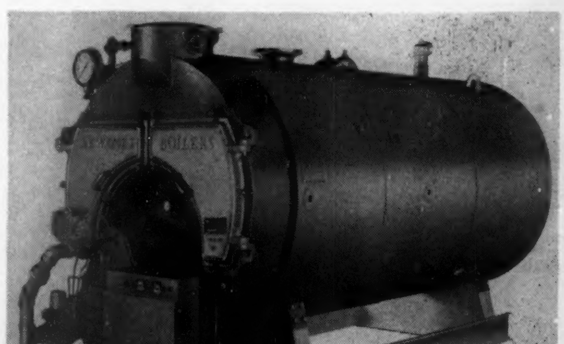
THERMOSTATIC REGISTER

Controlling heat in individual rooms, this newly introduced thermostatic register has a range of 63 to 77 F. Although the system for which it was developed is started and stopped by a centrally located main thermostat, room temperatures can be controlled by the register dial.

Jet-Heet, Inc., Englewood, N. J.

PACKAGED COMMERCIAL BOILERS

Supplementing the existing line of commercial packaged boilers, Scottie Junior units have output ratings of 20 to 150 hp for high pressure and 672,000 to 5,040,000 Btu/hr for low pressure. High pressure units



are designed for 125 to 150-psi steam, with higher pressures on application; low pressure units are de-

signed for 15-psi steam and 30-psi water, with higher water pressures available on application.

Forced draft operation is utilized to provide the combustion air necessary for stub stack applications. Firing rates are matched to heating surfaces and flame size and shape are adapted to furnace dimensions. Three sizes of Kewanee forced draft burners are matched to the new boilers to cover the entire capacity range. Three-way adjustment with an inlet louver, discharge damper and primary-secondary proportioning shutter results in maximum burning efficiency under most operating conditions.

All boilers are equipped with a range of standard features and equipment, including steel skid mounting, gas-tight front steel smokebox with gasketed cast iron doors, explosion relief doors, operating limit control, safety high limit control and insulated steel jacket with reinforced catwalk area.

American Radiator & Standard Sanitary Corporation, Industrial Div, Detroit 32, Mich.

ROOF-TOP COMBINATION UNIT

Combined in this roof-mounted heating, ventilating and air conditioning unit are two Landmark models which deliver either 7½ or 10 ton of cooling, with a heating capacity from 204,000 to 340,000 Btu/hr (in increments of 34,000 Btu/hr). Temperatures may be controlled by a single heating-cooling thermostat or by two thermostats, each controlling a separate

zone. Conditioned air may be ducted or distributed through a specially designed diffusing head which projects below the ceiling surface.

Lennox Industries, Inc., Marshalltown, Iowa.

UNDER-WINDOW ENCLOSURES

Offered are rigid metal panels serving as removable under-window enclosures for air diffusion units. Available in a variety of finishes, including furniture steel, wood veneer and stainless steel, the enclosures afford ease of installation, both on renovation work and new construction. Additionally, maintenance access problems are cited as being reduced greatly. The panels are flush, with no screws, locks or handles visible.

Buensod-Stacey, Inc., 45 W. 18th St., New York 11, New York.

EXPANSION JOINT

For use as a flexible connector in steam power plants between the turbine and the condenser, in duct systems and in similar large circumferential openings between equipment or systems, this endless Belt-Type expansion joint is molded of duck and rubber with rope-core beads integrally formed on each side of the web of the joint. The beads are inserted and tightened in clamping devices fastened to the equipment to be connected.

Standard Type GRC rubber expansion joint is 9⅜ in. overall width, with a web thickness of ⅝ in. and a bead thickness of 1¼ in. Circumferential length can be furnished to any specified length within a tolerance of ± 1 in. Joints have been made in excess of 100 ft in circumference. Maximum operating pressure is 15 psi or up to 30 in. Hg vacuum. Maximum working temperature is 180 F. Connector can take up to one in. compression and ½ in. lateral movement. It is not designed for extension; however, stretching of the unit for installation is allowed up to two in. per 35 ft of length and ⅛ in. per ten in. of width.

General Rubber Corporation, 66 Summit St., Tenafly, New Jersey.

AXIAL FANS

R-series axial fans, cited as being versatile large fans in both belt and direct drive, are available for such applications as cooling towers and exhaust units. These fans are offered in nine standard diam from 24 to 60 in. and in blade pitches of 20, 27 and 40 deg. Disc-hub-blade assembly has been designed for minimum stress and quiet operation. Availability of several pitches makes it possible to obtain high air delivery with extremely low fan axial depths.

Torrington Manufacturing Company, Air Impeller Div, Torrington, Conn.

AIR CONDITIONERS, HEAT PUMPS

Four new units have been added to broaden the coverage of this line in the commercial and industrial air conditioning and heat pump fields. Designated TA96A and TA120A (eight and ten-ton straight cooling units) and WTA96A and WTA120A (eight and ten-ton heat pumps), the new models feature an updraft design, with the fan being the only moving part exposed. Compressor, blower-motor and controls are protected in a separate compartment. Coil grille is of vertical design to discourage ice build-up during defrost on heat pump models and external sump heaters are provided to keep compressor oil

refrigerant-free for properly lubricated start. In addition, a heated, all-copper drainage system prevents ice build-up by circulating warm, liquid refrigerant through the drain pan to melt defrost slush.
General Electric Company, Central Air Conditioning Dept., Troup Highway, Tyler, Texas.

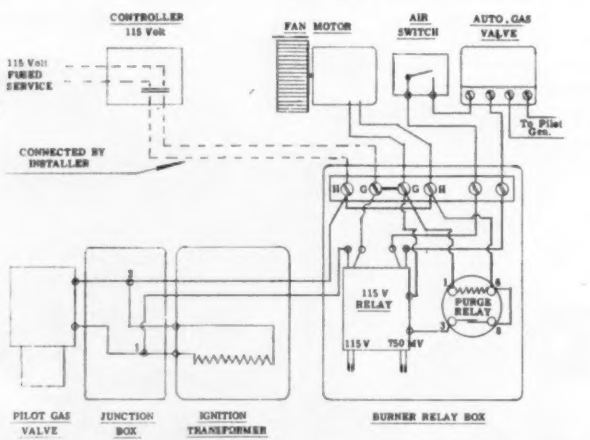
AIR DIFFUSERS

Permitting use of high cooling temperature differtials, ASL Architectural Straight Line Air Diffusers are slender in design, with symmetrical vanes. Suited for continuous wall-to-wall application, they also can be used as individual diffusers. Units are easy to install and require no screws, nuts or bolts.
Anemostat Corporation of America, 10 E. 39th St., New York 16, N. Y.

GAS BURNER CONTROL SYSTEM

Available with the Contractor line of power burners, rated from 75,000 to 400,000 Btu/hr, and FD-600 Power Burner, from 400,000 to 600,000 Btu/hr, the Ventronic system is shown here in a schematic diagram. Features cited are: one-min purge period before each burning cycle, continuous spark operation during burning cycle and automatic re-cycling of ignition.

As the thermostat or controller calls for heat, the burner blower starts and purges or vents the com-



bustion chamber for one min before the sequencing relay cuts in to energize the electrode, which releases a 6000-volt spark across the pilot. Then the pilot valve opens and the pilot ignites. In approximately 35 sec (in a cold chamber), the lighted pilot heats the thermocouple generator, providing sufficient power to open the main gas valve automatically.
Barber Manufacturing Company, 1052 E. 134th St., Cleveland 10, Ohio.

HIGH-CAPACITY VALVE

Greater capacity in a smaller size than offered previously is featured in a new high-performance expansion valve. Designed to meet industrial and commercial air conditioning and refrigeration requirements, the valve can be used in both original equipment and general replacement installations where space is at a premium. It is available in both internally and externally equalized and pressure-limiting types.

Used with Refrigerants 12 or 22, the 900 Compact Capacity Expansion Valve has a capacity in ton of 0.5 to 3 and 0.8 to 5, respectively. Small removable power element of the unit, of stainless steel construction, is shielded-arc welded. Valve body is a high density brass forging. A large inlet strainer, held in position by a retainer clip, is removable for cleaning.
American Radiator & Standard Sanitary Corporation, Controls Div, 5900 Trumbull Ave., Detroit 8, Mich.

SMALL ENVIRONMENTAL CHAMBER

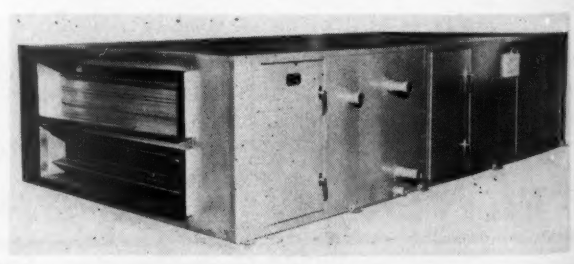
Whether used for transistor testing, miniature bearing manufacture or numerous other applications, the Penguin/5 is practical where space is at a premium. This 0.5-cu ft chamber occupies less than 2½ sq ft of floor space and is caster-mounted for portability. Standard low temperature range is from −10 to −60 F; unit is available with a high range to 500 F. The chamber is 15 in. long, 9 in. wide and 8 in. deep and is surrounded by 2½ in. of low K factor insulation.
Cincinnati Sub Zero Products, 3932 Reading Rd., Cincinnati 29, Ohio.

INSULATING CEMENT

Consisting of a special mixture of mineral wool and diatomaceous earth, SR is a new insulating cement blended to form a wall of dead air cells that stops heat transmission. Light and strong, the cement adheres to metal, tile, brick, insulating block and blankets, and is cited as staying on at temperatures to 1800 F. Application instructions call for mixing with water and troweling or applying with bare hands. The material dries to a hard finish with little shrinkage.
York Insulation Company, Hillside, N. J.

AIR FILTRATION UNIT

For supply of heating, ventilating, filtering and cooling needs to dairy and food processing plants and laboratories, the Klimatic King Series 70, a high-efficiency air filtration unit, removes air-borne bacteria, reduces excessive humidity and maintains desired room temperature. Air passing through the unit is filtered twice: through a standard washable type



filter to remove larger dirt particles and a high efficiency dry type filter that is cited as having a cleaning efficiency comparable to electronic precipitators.
King Company, 121 W. Winona St., Owatonna, Minn.

News highlights of the month

TRENDS

Thermoelectric water cooler

Using the principle of thermoelectricity to cool water electrically without the conventional compressor and refrigerant, thermoelectric water coolers have been announced by two manufacturers. Westinghouse has begun production of a bottle-type cooler. General Electric has introduced a pilot cooler, but it is stated that production plans for the new unit would depend upon market test results. Norge Div, Borg-Warner has announced production-line output of ice-cube freezers (ASHRAE JOURNAL, April 1961, page 21).

Thickness of insulation

Announcement of a new system for specifying exact thickness of thermal insulation needed for maximum economy, developed by the Union Carbide Chemicals Co. in cooperation with the College of Engineering, West Virginia University, was made on March 24 at a meeting sponsored by the National Insulation Manufacturers Association. By referring to charts, instead of working out complex mathematical formulas, an engineer can find the solution for any possible combination of variables, involving cost of heat energy, cost of insulation, cost of capital at various periods of amortization and conductivities of various insulations at different temperatures. Although the basic equations for the calculation of economic thickness of insulation were worked out in 1926 by L. B. McMillan to ASME, they were found difficult to apply. The new method combines into a single "D" factor the variables involved in heat cost and insulation cost and the formula is made workable by means of a series of four sets of charts. NIMA will publish the formulas and tables in a revised edition to the "Manual on Economic Thickness of Insulation for Flat Surfaces and Pipes" for general distribution to industry.

A new No. 2 oil

Changes in gravity requirements of No. 2 Heating Oil have been recommended by Technical Committee "E" of Committee D-2 of the American Society for Testing Materials. These changes, which are subject to confirmation by mail ballot by the overall committee and ASTM, supposedly will result in greater uniformity of product and consequent improvement in the performance of oil burning equipment.

Air conditioners gain

As released by Air-Conditioning and Refrigeration Institute, figures for total 1960 shipments of unitary air conditioners (not including room units) show a gain of about 9 per cent over 1959 shipments. It is estimated that the units reported to ARI represent more than 90 per cent of the industry total.

BOOK REVIEWS

Higher salaries

Salary levels of engineers rose approximately 5 per cent per year between 1958 and 1960; overall median salary now stands at $9,600. These findings are revealed in the Engineering Manpower Commission survey "Professional Income of Engineers—1960", which covers approximately 200,000 engineering graduates in industry, education and government. Copies of the report are available from EMC, 29 West 39th Street, New York 18, N. Y. at $3.

Operation Cryogenics

Suitability of 9 per cent nickel steel vessels for service applications down to −320 F was tested in a program conducted jointly by the International Nickel Company, Chicago Bridge and Iron Company and United States Steel Corp. Complete data from these tests are included within the pamphlet, "Final Results from Operation Cryogenics". A smaller booklet, "Questions and Answers from Operation Cryogenics", which outlines mechanical properties and fabrication of the cryogenic vessels, is inserted within the pamphlet.

Multi-lingual dictionary

Published in six languages (English, French, German, Russian, Spanish and Italian), the *International Dictionary of Refrigeration* contains 1600 expressions, of which 400 are defined, within 300 pages. Established by the International Institute of Refrigeration, 177, Boulevard Malesherbes, Paris 17, France, in cooperation with technical societies from each of those countries, including ASHRAE. Dictionary comprises two parts – one contains all index numbers classified methodically, each identified by a numerical index, corresponding with the six languages; the second part includes a series of six alphabetical indexes, one for each language, referring to the arrangement of the first part. Price is $6 if ordered before May 31; special rate for Associate Members of IIR is $4.50.

API Proceedings Now available, Proceedings of the API Research Conference on Distillate Fuel Combustion includes texts of papers presented at that Conference, held in Chicago, Ill., March 14 and 15, covering research and development on oil burning equipment and components. Copies may be obtained from the American Petroleum Institute, 1271 Avenue of the Americas, New York 20, N. Y. for $5.

Cryogenics Included within Volume 6 of *Advances in Cryogenic Engineering* are 65 technical papers which were presented at the 1960 Cryogenic Engineering Conference, sponsored by the University of Colorado and the National Bureau of Standards. Edited by K. D. Timmerhaus, the illustrated 652-page book may be ordered from Plenum Press, Inc., 227 West 17th Street, New York 11, N. Y.; price is $15 (domestic) and $17.50 (foreign). Volumes 1-5 of this series are also available from the publisher.

Welded steel tubing "Handbook of Welded Steel Tubing" includes latest engineering and design data in the manufacture and fabrication of electric resistance welded carbon and stainless steel tubing. Published by the Welded Steel Tube Institute, this comprehensive handbook is designed to update all technical and manufacturing information on the subject. Available from the Institute, 1604 Hanna Building, Cleveland 15, Ohio, price is $10.

Refrigerant piping Now ready for publication, "Refrigerant Piping Data" contains information in tabular and chart form not available elsewhere, together with a referenced discussion of the basis for refrigerant piping design including instructions, with worked-out examples for application of the tables and charts. Prepared by Air-Conditioning and Refrigeration Institute (1346 Connecticut Avenue, N.W., Washington 6, D. C.), copies of the 64-page booklet may be ordered now, at $3 each, in order to determine the print run.

Boilers Of interest to those responsible for the design, installation and operation of equipment for generating heat and power, *Boilers: Types, Characteristics and Functions,* by Carl D. Shields (member ASHRAE) is a practical engineering approach to the selection, application and performance of boilers. Well illustrated, the book contains 32 chapters and 559 pages. Published by F. W. Dodge Corp., 119 West 40th Street, New York 18, N. Y.; price is $15.

SPECIAL MEETINGS

BRI to meet As part of its 1961 Spring Conferences, to be held in Washington, D. C., May 16-18, the Building Research Institute has scheduled a comprehensive three-part program on adhesives and sealants. Sessions will be held on: Requirements for Weatherproofing Thin Shell Concrete Roofs; Pressure Sensitive Tapes for Buildings; and Selection and Field Application of Adhesives. Also on the program is a session covering Reports on New Research and Suggestions for Further Study, with ASHRAE Presidential Member and Fellow P. B. Gordon, chairman of the BRI Research Committee, as chairman.

London meeting Second International Refrigeration and Air-Conditioning Exhibition and Convention was held in London on April 11-14. Exhibition featured machinery and equipment for virtually every kind of refrigeration, air conditioning and mechanical cooling application. Speakers at the Convention included ASHRAE members F. L. Levy, A. K. Alcock and W. E. MacMillan.

Environment study "To explore environmental resources and their value and usefulness for man in terms of needs for industry, government and conservation–in keeping with health, social and economic trends" and "To consider principles and methods of practice which will encourage a maximum development of our environmental resources for society" – these are the stated purposes of the Second National Congress on Environmental Health, sponsored by the University of Michigan School of Public Health, the National Sanitation Foundation and the American Public Health Association, to be held in Ann Arbor, Mich., June 6-8.

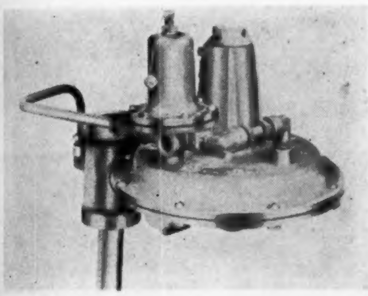
INDUSTRY CHANGES

New name National Oil Fuel Institute is the name of the new national oil heat association formed by the merger of the Oil-Heat Institute of America and the National Fueloil Council (ASHRAE JOURNAL, March 1961, p. 22). To develop increased programs for all segments of the industry, the new organization plans to continue and expand constructive activities of both OHI and NFC.

PARTS AND PRODUCTS

PILOT, REGULATOR

Combining a cast iron, high pressure, Model 173 regulator assembly and standard diaphragm with a Model 080R pilot, this unit has an operating range of 10 to 100 psig inlet pressure, 2 to 20 psig outlet pressure. Capacity



ranges from 2000 to 15,000 cfm of 0.6 gas, dependent upon pressure and orifice size combination.

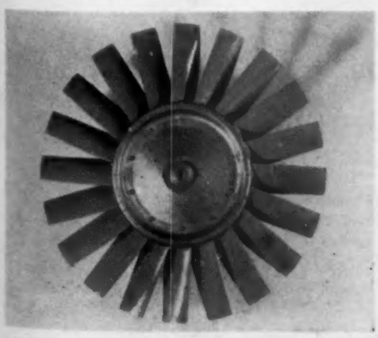
Constant pilot loading is cited as almost eliminating spring effects and diminishing the diaphragm effect. Internal relief for pressure build-up within the pilot-loaded upper case of the regulator is provided in the pilot. Since this is an intermittent relieving system and vents to the atmosphere, the installation must be compatible with this type of regulator.

Unit is available in one x one-in. standard and ¾ x ¾-in. optional connection sizes. Standard orifice size is ¼ in. with optional orifice sizes of 3/16, 3/8 and 1/2 in.

Rockwell Manufacturing Company, Meter and Valve Div, 390 N. Lexington Ave., Pittsburgh 8, Pa.

AXIAL FLOW FANS

Actually a system of 72 different fans assembled from modular units, the Multi-Wing axial flow fan line offers



five sizes of hub and two sizes of airfoil blades pitched 30, 35, 40 and 45 deg. Overall diams range from 10 to 28 in. Blades are of high density polyester or polypropylene plastics adapted to a wide range of working temperatures and safe-rated to more than 4000 rpm.

An installation can be modified simply by disassembling the fan and inserting different blades. In some cases, where high static pressures are no problem, power consumption can be reduced by halving the number of blades and using spacers between.

Mainstream Company, 1323 First St., Coronado, Calif.

ELECTRONIC AIR CLEANER

Available in a portable, plug-in floor model and a suspended ceiling model, this self-contained package unit for use in restaurants, retail stores and

laboratory and hospital room areas is cited as cleaning, deodorizing and recirculating air in rooms up to 5000 cu ft. Combined in the unit are the high dirt and smoke removal efficiency of an electronic air cleaner and an activated charcoal after-filter for removal of odors. Included is a 1000-cfm blower.

Electro-air Cleaner Company, Inc., Olivia & Sproul Sts., McKees Rocks, Penna.

GAS AREA HEATERS

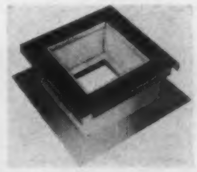
Twelve models – a B and C series, each of which has six models in a range of 35,000, 50,000 and 65,000 Btu/hr – comprise the Baronet Series of vented gas area heaters. The two series vary according to standard and optional equipment and use of natural and LP gas.

Featured on the 35,000 and 50,000-Btu/hr models is the Hi-Crown burner; 65,000-Btu/hr units have a slotted burner. Other features include Pyrex glass fronts, built-in draft diverter and a rounded base flange to protect the floor. Standard on the B Series is a control center mounted on the top rear of the cabinet. Optional is the Thermothrust blower, with quiet motor and rubber mountings.

Dearborn Stove Company, 1700 W. Commerce, Dallas 22, Texas.

FAN CURB

Cited as eliminating improvised field construction for installing roof exhaust fans, the Thermal-Acoustic Curb provides thermal insulation to prevent rusting and rotting damage to a building. Spun glass fiber absorbs noise and reduces reverberation. All-aluminum, welded construction eliminates joints and a strong fastening flange is provided for duct connection.

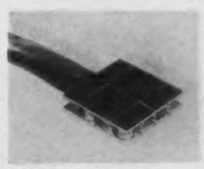


Davidson Fan Company, 213 California St., Newton 58, Mass.

THERMOELECTRIC UNITS

Overheating of black box circuits, caused by heat dissipation of power transistors and other components, is cited as being corrected by various spot cooling applications of new thermoelectric components. Currently introduced is the TA-20, a Thermo-Array of several smaller TA-12 junctions, also produced as a finished Thermo-Module, the TA-20M, with 1/16-in. aluminum plates on top and bottom, ready for use as a complete thermoelectric unit.

Small in size (1½ x 1½ x ½ in.), the module can pump more than 20 watt or attain a temperature differential between hot and cold plates in excess of 80 C. In cases where several individual modules are connected in series or parallel, the capacities attained are even greater.



Ohio Semiconductors Div, Tecumseh Products Company, 1205 Chesapeake Ave., Columbus 12, Ohio.

SLOT-TYPE DIFFUSERS

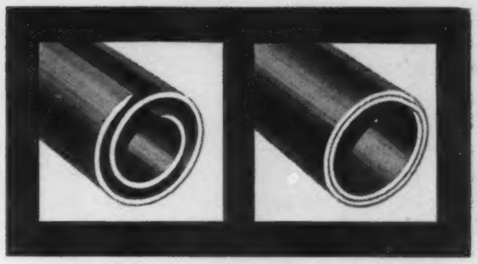
Extruded aluminum, Stripline slot-type air diffusers, with separate plaster frames and removable cores for heating and cooling applications, are

ing

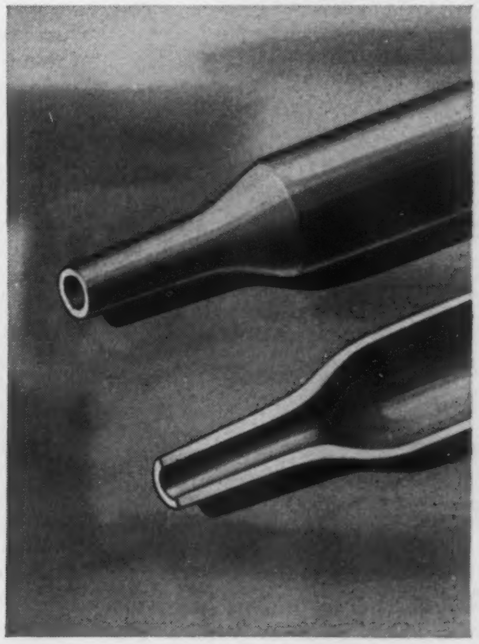
nd this mass-fabrication experience can help you. Whatever kind f small-diameter steel tubing you are now using—or plan to use—undy can probably make it better . . . in quantities of thousands r millions. How? By making your part from Bundyweld®, he steel tubing that's rated tops in quality in the refrigeration ndustry. Bundyweld meets ASTM-254; and Govt. Spec. MIL-T-520, Type III. And by applying years of problem-solving experince to your tubing problem, Bundy engineers can often find short uts that shave costs in mass-fabrication. Let Bundy experience elp you. Write: Bundy Tubing Company, Detroit 14, Michigan.

UNDY TUBING COMPANY • DETROIT 14, MICH. • WINCHESTER, KY. • HOMETOWN, PA.

Vorld's largest producer of refrigeration tubing. Affiliated plants n Australia, Brazil, England, France, Germany, Italy, Japan.



Bundyweld, double-walled from a single copper-plated steel strip, is metallurgically bonded through 360° of wall contact. It is lightweight and easily fabricated . . . has remarkably high bursting and fatigue strengths. Sizes available up to ⅝" O. D.



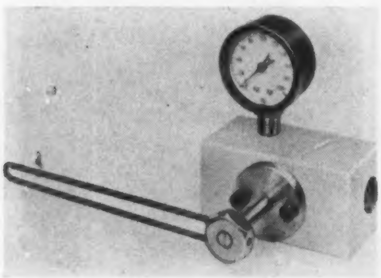
Bundyweld tubing can easily be swaged down to capillary size with the inside diameter held to very close tolerances. Swaging the end eliminates an additional joint. The ⅜" Bundyweld tubing shown above is swaged down to .081/.084 I.D.

BUNDYWELD® TUBING

available in two styles of frames and three types of cores with diffusing vanes for ceiling, sidewall and window-sill installations. Diffusers are made in sections or can be used as a continuous unit by butting the sections together with a built-in interlocking feature. Use of screws and screwholes is eliminated by a coil spring-lock.
Air Devices, Inc., 185 Madison Ave., New York, N. Y.

OIL VALVE

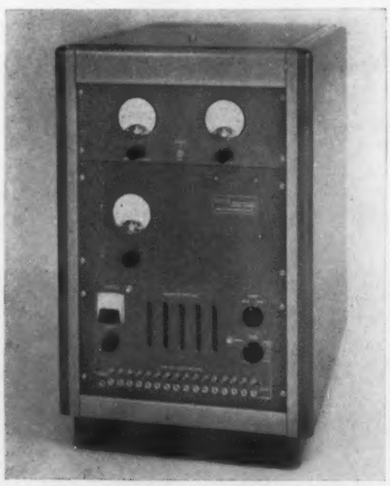
Engineered for No. 4, 5 and 6 oils, the 4-P oil valve is cited as eliminating poor combustion, excessive smoke and soot formation, poor modulation, frequent adjustment, excessive maintenance and excessive pre-



heating of oil. Body of the unit is a solid piece of machined brass, and the valve has but one moving part to assure trouble-free operation.
Dawson Enterprises, 631 Central Ave., East Orange, N. J.

DILUTION SYSTEM

By the addition of an air-dilution system to Model PC200A Airborne-Particle Monitor, this manufacturer is cited as providing an instrument ca-



pable of describing accurately the spectrum of particles present in specimens of air ranging from clean-room atmospheres to highly polluted industrial smogs. In addition, the dilution system, available by itself, can be used in a laboratory to give an accurate means of mixing air or gases in controllable proportions.

Used with the particle monitor, the dilution system mounts in a standard 19-in. rack or in a metal cabinet 24 in. wide by 30 in. deep by 38 in. high as shown. Provided is a continuously variable addition of filtered dilution air from zero to 200 cc per min to a basic sample, giving dilution ratios from zero to 50:1.
Royco Instruments, Inc., 440 Olive St., Palo Alto, Calif.

MOTOR ACTUATORS

For positioning air dampers, control valves, programming devices, burner fuel valves and similar equipment, these motor actuators are available in a wide variety of models. Units are offered with various operating speeds and output torques to suit many applications. Models are available which incorporate an additional balancing potentiometer for simultaneous operation of a second motor actuator.

Dust-tight and splash-proof, Type MA units are powered by a split-phase capacitator-type motor, which drives the output shaft through precision hobbed reduction gears. Cam-operated limit switches stop the actuator movement at predetermined travel limits. Proportional motor actuators include a precision wound potentiometer which is positioned directly from the output shaft to provide a variable voltage signal indicative of actuator position. Drive motor and gear train are sealed in oil, providing permanent lubrication, maximum cooling and minimum maintenance.

Units operate on 24-volt ac, 50-60 cycle. Control circuit is three wire, 24 volt. Proportional actuators use 135-ohm, two-watt potentiometers.
Penn Controls, Inc., Goshen, Ind.

CENTRIFUGAL FANS

Small direct-connected, belted and portable models in Types BG and FC Util-A-Vent Jr. series, as well as Type GP volume units and Type GPE pressure exhaust centrifugal fans are offered by this manufacturer.

Type BG Util-A-Vent direct-drive models are available in five sizes and 14 capacities for supply and exhaust applications where low pressure, small volume, quiet operations are desirable. Type FC belted units are offered in two models with adjustable V-belt drives for applications where speed adjustments are required. Gasoline engine-driven and portable models are also a part of the BG line.

Available in four sizes and nine capacities, the Type GP centrifugal fan is designed for exhausting fumes, handling light dust or any small exhaust purpose. It can be built into other machines.

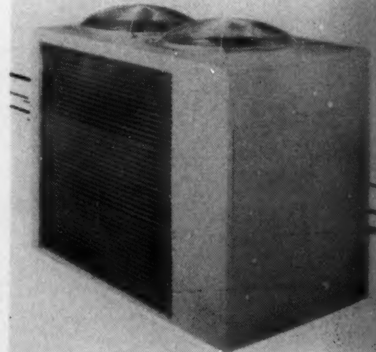
Type GPE pressure exhaust fans are suitable for a wide variety of applications, including dust removal, conveyance, exhaust, supply blast and air supply. They are available in four direct-connected and three belted sizes.
General Blower Company, a subsidiary of Ilg Electric Ventilating Company, Morton Grove, Ill.

CIRCULATING HEATERS

Each of the six new Brilliant Fire circulating gas heaters is available in two series, MTW (Fire Light) and MOT (Deluxe Console), in either 35,000, 50,000 or 56,000-Btu/hr sizes. All units are equipped with a factory installed and connected combination safety valve and thermostat, with the thermostat knob located at the top left corner of the cabinet. Features are a large front service door for easy access, all-weather draft diverter and a lint-proof pilot.
Ohio Foundry & Manufacturing Company, P. O. Box 191, Slack St., Steubenville, Ohio.

AIR COOLED UNITS

Introduced for split system installations are ten-hp Model 1212 and fifteen-hp Model 1216. Capacity of the former, when used with a Model 1412 cooling coil (which delivers



4000 cfm of air), is 114,000 Btu/hr. Model 1216 can be matched with Model 1416 cooling coil, which re-

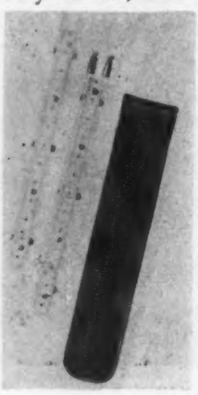
sults in a capacity of 154,000 Btu/hr and 6000 cfm, respectively. Paired with two Model 1489 cooling coils, the 1216 delivers 166,000 Btu/hr.

Each unit uses a five-cylinder compressor, operates at 120 F outside temperature and will run when voltage has been reduced as much as 10%. All safety controls are reset at the thermostat. Each has controls to assure operation down to 55 F outside temperature and accessory cold weather controls permit operation down to 0 F by dampering air and not overcharging the unit with refrigerant. A separate receiver and strainer-drier are standard, as is a hot gas by-pass which permits light load operation without coil freeze-up.

Chrysler Corporation, Airtemp Div, P. O. Box 1037, Dayton 1, Ohio.

MANOMETER

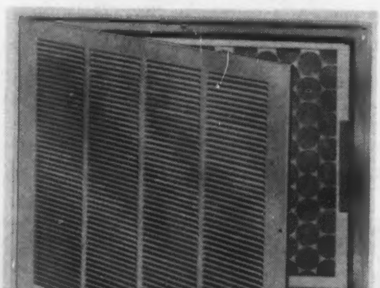
Suitable for water, glycerine, mercury or oil, this new manometer is offered in two models, open-ended and valved. The latter, with two brass needle valves, permits fluid retention if desired, and allows readings to be taken by unscrewing the needle valve a half turn. Units are available in five sizes, from 6 to 18 in. Standard on all models is ⅜-in. plastics tube.



Gas Consumers Service, 45-46 21st St., Long Island City 1, N. Y.

FILTER GRILLE

Especially suited for installations where heating or air conditioning units are located in attics, crawl spaces



or other inaccessible areas, this new grille, with a hinged frame, has face bars stamped ½ in. wide on a ½-in. center and a 30-deg deflection. In addition to providing greater free area, the face bars eliminate visibility of filter surface and permit installation with deflection either up or down. Filters are recessed approximately one in. behind the grille face to provide greater utilization of filter surface. All standard one-in. permanent or replaceable type filters can be used.

Auer Register Company, Cleveland, Ohio.

HEAT PUMP

Attaching directly to an outside wall, this new heat pump is suited to many

applications. Rated at 18,000 Btu/hr, each unit can air condition 1000 sq ft and may be used in multi-storied dwellings with one unit serving each level. Installation is possible from a balcony, porch or garage roof. A thermostat is built-in and the unit is equipped with twin squirrel-cage blowers, driven by a ⅛-hp motor and capable of moving 650 cfm of air through the ductwork.

Units are of the air-to-air type, drawing heat from the air inside the structure and transferring it to outside air during the summer. In winter the process is reversed.

Westinghouse Electric Corporation, 3 Gateway Ctr., Pittsburgh 22, Pa.

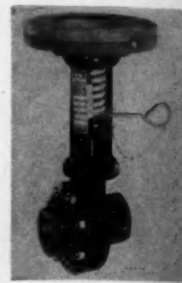
REMOTE CONDENSER

Arrangement of coils in a V-shape so that air is drawn through them and discharged vertically upwards by the propeller fans is cited as providing a low silhouette in a minimum of floor space in this remote, air-cooled condenser, the V-Con. Other features include direct-drive fans, totally enclosed motors and lightweight, corrosion-resistant, aluminum cabinets. Initially, the V-Cons are being offered in capacities of three to ten ton.

Halstead & Mitchell, Inc., Bessemer Bldg., Pittsburgh 22, Pa.

CONTROL VALVES

Now available in stainless steel as Model 770 and in cast steel as 670, control valves are manufactured in ¼, ⅜, ½, ¾, 1 and 1¼-in. capacities. Stainless steel trim is standard with linear flow characteristic sliding gate seats. Suitable for steam, air, water, oil and gas, valves are either self or controller-operated, and are offered for 300 psi and 500 F WSP service.



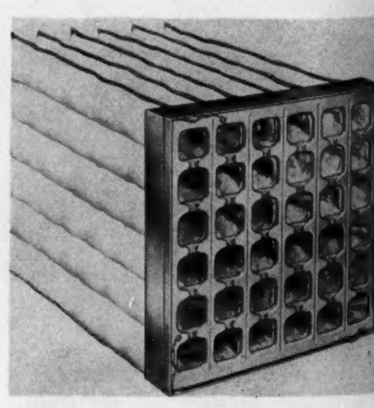
OPW-Jordan Corporation, 6013 Wiehe Rd., Cincinnati 13, Ohio.

DRY-TYPE FILTER

Air cleaning efficiencies of the Dri-Pak air filter are cited as being as high as 97%, based on the Dust Spot Method on Atmospheric Dust. Offering low resistance, compactness and excellent dust-holding capacity, the filter is suited for use in industrial and commercial ventilating systems or in central air conditioning systems.

Inflating when the system is in operation and collapsing when the system is shut down, the unit can be inspected and serviced from either the air-entering or air-leaving side of the filter bank. Since no back-up wire grid is necessary, chances of damaging the filter during installation have been reduced greatly.

The filter is disposable, light in weight and requires little storage space. Frame is available in three

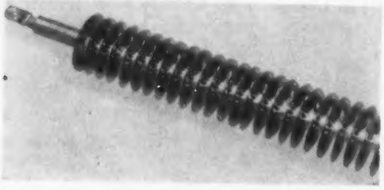


arrangements: front access with and without pre-filter and rear access with no pre-filter. Each 24 x 24 x 36-in. filter is rated at 2000 cfm and provides 95 sq ft of filtering media sur-

face. Velocity through the media is but 21 fpm and the ratio of media area to face area is 24:1.
American Air Filter Company, Inc., 215 Central Ave., Louisville 8, Ky.

TUBULAR ELEMENTS

In rating sizes up to 350 watt per ft in convection and 1000 watt per ft with fan cooling, this line of noiseless,



finned, tubular heating elements has been designed for fan heater, blower coil and convection baseboard applications. Continuous lengths up to 12 ft are available with copper-hydrogen brazed steel fins spaced according to application requirements. Standard spacing is 3½ fins per in.; spacing from three to eight per in. can be supplied.
Still-Man Manufacturing Corporation, 429 E. 164th St., New York 56, N. Y.

ROOM UNIT

Designated Cavalier, this 230-volt unit delivers 9600 Btu/hr and is but 14 in. deep, 16 in. high and 26½ in. wide. A single airflow control permits air to be directed straight into a

room or upward for draftless circulation. Quiet operation is cited as being achieved by a new radiax fan, resilient rubber fan mountings and internal blanketing with glass fiber insulation. The unit can be taken easily from the window without removing the sealed-in mounting cradle.
Carrier Air Conditioning Company, Syracuse 1, N. Y.

AIR HANDLING LINE

New products offered include a twin-duct air blender, packaged multizone unit, Roofair for roof-mounted cooling or heating and a line of ventilating fans. Series 600 Centriflow Fans feature a wheel design with backwardly inclined blades to assure smooth and unrestricted air flow for high efficiency and quiet operation.

Designed for high velocity, double-duct air conditioning systems, the air blender is adaptable for ceiling and under-window installation. The blenders operate without the use of motors, piston separators or mechanical linkage.

Three models, having a cooling capacity range of 20 to 30 ton, comprise the Multitemp CMC Series, packaged multizone unit line. Heating coils are provided with heating zone controls for heating, cooling or a combination of both.

Cooling and dehumidification are provided with the basic WRA Roofair unit, which also is available with gas, oil or steam heat for year-round conditioning.
Worthington Corporation, Harrison, New Jersey.

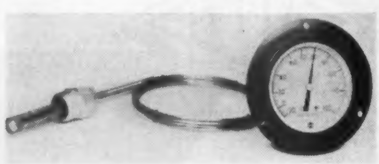
SOLENOID VALVE

For use in liquid and gas control, this compact, straight-through flow, high pressure solenoid valve is available in 6 to 64-volt dc; 6, 12, 24 and 110-220-volt, 60-cycle ac; and 110-volt, 400-cycle ac models. Unit operates in any position with a maximum rate of 1000 cycle per min at 100 psi. It is available also in water and fungus-resistant models.

Using from one to three watt of power, the VO46A valve will serve under pressures to 200 psi in temperatures ranging from −65 to 350 F. Inlet connection is ½-in. NPT with an outlet connection of ⅛-in. NPT. Self-supporting, in-line mounting is featured. Major parts of the unit are stainless steel and are silver-brazed for rigidity. No gaskets are used.
General Magnetics, Inc., 2641 S. Louisiana Ave., Minneapolis 26, Minn.

THERMOMETERS

Serving as remote temperature indicators, vapor tension-actuated equip-



ment thermometers, Series 8000, are center back connected and panel mounted, but may be designed for individual applications. Dial sizes are 2, 2½ and 3½ in., with dial markings unevenly graduated because of the nonlinear relationship between the temperature and pressure of the actuating medium. Bourdon tube is made from non-ferrous metal and the newly designed movement is composed of corrosion resistant materials.
American Machine & Metals, Inc., U. S. Gauge Div, Sellersville, Pa.

LIQUID LEVEL GAUGE

Visual indication is utilized by the Gage-O-Matic to provide constant information as to the status and behavior of liquid in a refrigerant vessel. In addition, the unit is manufactured to order, with single or multiple control switches to provide automatic operation of pumps, valves, indicator lights, alarm bells, compressor shutdown and signal circuits to protective organizations.

But five major parts comprise the control: float chamber, guard section, control switch or switches, combination float ball, rod and magnet-indicator and an indicator glass. Control switches are located away from the cold influence of the refrigerant, but are so constructed that they will continue to function even if covered by frost or ice.
H. H. McKinnies Company, 3131 W. Mill Rd., Milwaukee, Wisc.

REMOTE CONDENSERS

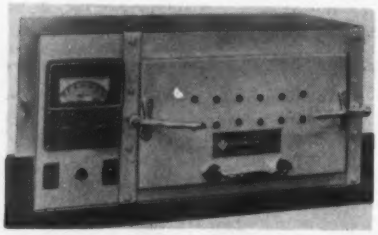
In six models, air-cooled condensers are suited for use in such applications as built-up air conditioning systems, package air conditioners, refrigeration systems in which multiple compressors are connected to single multi-circuit condensers and commercial heat pumps. Units are adaptable to new construction and can be used also to convert from water to air-cooled systems. An optional feature is Limitizer head pressure control, which makes it possible to maintain proper operation throughout the year, regardless of weather conditions.

Models RVD and RHD are designed to meet requirements of both the air conditioning and commercial refrigeration fields. Vertical air discharge units are suited for location in outdoor areas where low vertical projection of the equipment is desired and horizontal discharge units are

adapted best to installation in areas where conservation of floor or roof space is necessary.
Bohn Aluminum & Brass Corporation, Danville Div, Danville, Ill.

THERMAL SHOCK CHAMBERS

Designed for bench top operation, the Ultratemp/600 and Ultratemp/



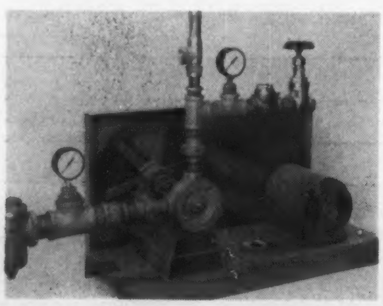
800 have a net work area of ½ cu ft. Access to the steel chamber, which measures 16 in. long, 8 in. wide and 8 in. high, is by a removable, file-drawer type door. Model 600 is instrumented for operation between −100 and 500 F with a five-min pull-down and thirty-min warmup. Model 800 has a low temperature of −320 F and a high of 500 F. Each is instrumented to an accuracy of ± ½ F and protected by three in. of low K factor insulation.
Cincinnati Sub Zero Products, 3932 Reading Rd., Cincinnati 29, Ohio.

GLASS FIBER VENTILATORS

Molded glass fiber intake and relief ventilators feature low profile, weather resistance and flexibility. Throat-opening sizes range from 10 to 60 in., either square or rectangular. Accessories such as bird screens and various types of dampers are available also.
Williams-Bermuda Corporation, 310 N. Normandie Ave., Los Angeles, Calif.

FUEL PUMP SET

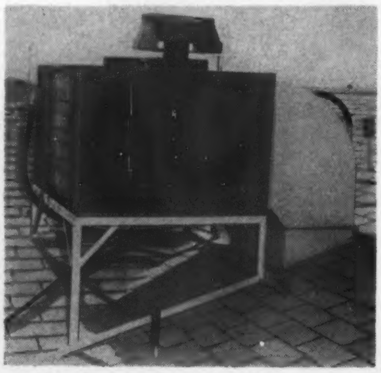
Containing the accessories required to perform the multiple functions of fuel transfer, pressure boosting and



fuel straining in one operation, a packaged pump set for industrial fuel oils is offered now. Included in the standard single set is a low speed rotary pump with a 2:1 ratio of fuel pumping to fuel burning for given boiler hp, single or double V-belt drive, basket-type single or duplex suction strainer, vacuum gauge, return-to-tank relief valve, discharge check and gate valves and pressure gauge. Pump speeds are from 450 to 550 rpm and the unit pumps from 45 to 100 gph, depending upon the motor selected.
Walter H. Eagan Company, Inc., 2336 Fairmount Ave., Philadelphia 30, Pa.

YEAR-ROUND UNIT

Combined in the Duopac are a horizontal-type forced-air gas furnace and an air-cooled, electric remote air conditioner, in one compact unit designed for outdoor installation. Ground level installations are located on a



slab, as in the case of a remote compressor. This slab is placed close to the sidewall of a house, with short connections to air delivery and return air ducts. Cooling coil is in the unit itself. In some applications, rooftop (as shown) or wall-hung installations may be used with attic distribution systems.

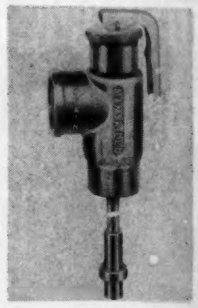
At the time of introduction, the Duopac is offered in two sizes, with a third to follow soon. Available now are Models 24/55 (two-ton cooling capacity, 55,000-Btu/hr heating capacity) and 36/80 (cooling capacity of three ton, 80,000 Btu/hr heating capacity).
Day & Night Manufacturing Company, P. O. Box 2222, La Puente, Calif.

MAKE-UP AIR UNITS

Capacities of the nine sizes in the Sun-Flo Model AH-2 series range from 20,000 to 60,000 cfm for air and from 1,875,000 to 6,000,000 Btu/hr. Each of the horizontal, dual-fan, direct gas-fired units is designed to maintain a constant discharge temperature of 80 F through a range of outdoor temperatures from −15 to 80 F. Controls are automatic.
Metals Engineering and Manufacturing Company, Inc., 8824 Lyndon, Detroit 38, Mich.

RELIEF VALVE

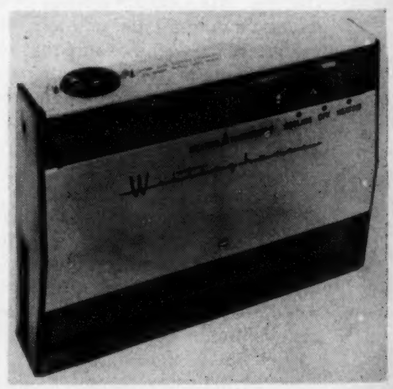
Operating between 200 and 210 F, the 503 automatically resets when temperatures return to normal. Features of the valve include heavy red brass body, test lever, stainless steel control spring, six-in. exposed extension tube and special bibb washer. In addition, over-



sized 8/16-in. seat is cited as providing instant relief and 3/16-in. flow passage as preventing liming or possible obstruction. Standard pressure setting is 125 psi. Steam rating is 80,000 Btu/hr.
Mansfield Sanitary, Inc., Perrysville, Ohio.

HEATING EQUIPMENT

New products added to this line of residential electric heating equipment include a three-ft long baseboard section rated at 750 watt and a combination thermostat-switch-outlet section. Incorporated in the three-ft section are advantages offered previously with two-ft sections: a modular plug-in concept that simplifies installation



time, reduced over-all size and three-way air flow design. This section is available in 120, 208, 240 or 277-volt and provides 2560 Btu/hr.

Nine and a half in. long, the combination section contains a standard heating thermostat and a 220-volt outlet for use with room air conditioners. For cooling, rating is 12 amp; the heat control is rated at 20 amp.
Westinghouse Electric Corporation, Box 2278, Pittsburgh 30, Pa.

JOURNAL
OFFICIAL PUBLICATION
MAY 1961

Comment

WHY DON'T YOU, HE SAID

"Why", asked a visitor, after expressing some generalized approvals and compliments, "are your Comments so often directed to the higher levels of interest of the members of our Society? Don't you have any message for those less professional?" Ordinarily we tend to skip isolated opinions as such, but our visitor raised several points of some significance and deserving of note, of which he was, no doubt, not really aware.

First, we do not conceive it to be any part of our function to be a Torch Bearer. We have no messages. We have nothing to prove. We espouse no causes. Our observations are solely the personal ones of a single looker-on. Again, we are not aware of making other than infrequent references to anything except the overall problems of people in general and of those belonging to a professional engineering society in particular.

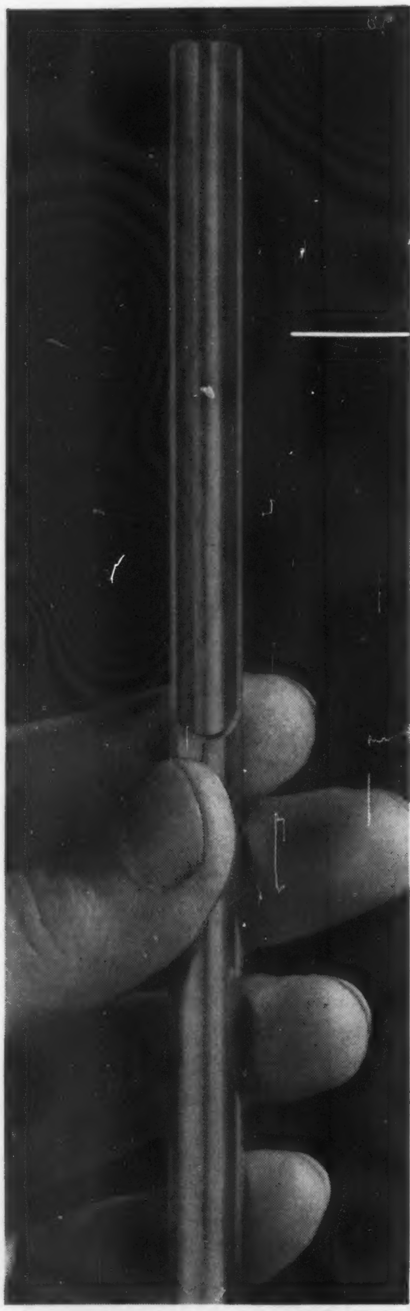
We do try to induce people to think for themselves.

On the other hand, we do have a conviction that allies with such fundamental things as that in this world each of us must grow or die. That we must continue learning or stagnate. That we must try to do a better job, more earnestly today and tomorrow than we did yesterday, or we shall end up by doing a poorer one.

More specifically, we think that any member of a professional engineering society should be forever trying to expand his personal knowledge of the field to which he has dedicated his efforts; should make the individual struggle to master aspects which elude him by reasons of inadequate training or forgetfulness of that which he once studied; should, in other words, really be an engineer.

We think that this is highly pertinent to matters of Society membership and we endeavor to perform our chores as Editor with that forever in mind.

Edward R Searles
Editor



23,000 miles of Anaconda thin-wall copper tube passed by eddy current test–

with no customer rejects

To meet rigid requirements for thin-wall copper tube for air conditioners consistently, Anaconda developed new techniques and standards for nondestructive testing that are highly reliable in flaw detection.

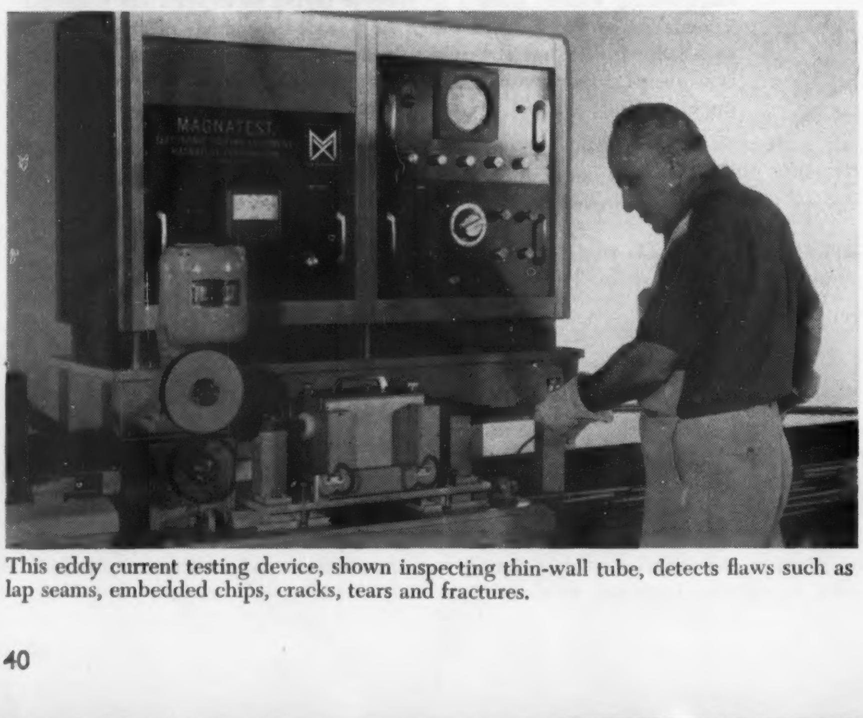
Eddy current equipment is Anaconda's inspection tool. It can be operated at many levels of sensitivity. Merely passing tube through this electronic device means nothing. It is only when used at a practical level for each size tube—with meaningful standards—that this type of testing is significant.

Anaconda has developed these critical and realistic standards through some 10 years of research in electronic testing. Substitution of automatic electronic inspection for human observation and judgment means uniformly excellent products for Anaconda customers.

Anaconda standards are based on a practical measure of tube soundness in actual use and end product. The eddy current device detects such flaws as lap seams, embedded chips, cracks, tears, and fractures—and rejects unsound tube automatically.

The care and attention Anaconda has devoted to developing electronic testing standards for tube used in air conditioning is typical of quality conscious practices throughout – for all types of copper tube used in air conditioning and refrigeration. It is your assurance of consistent soundness and dimensional accuracy in whatever Anaconda product you buy.

Quality tube and creative technical services. Whether you need hard or soft thin-wall tube, hairpin bends, long coils, capillary or restrictor tube, or one-piece bulb and capillary units, Anaconda has the experience and facilities to produce the quality tube you need for economical manufacture. For creative technical assistance in tube or tubing components, the services of Anaconda Small Tube Division specialists are available to you for the asking. For further information or technical help, write: Anaconda American Brass Company, Box 1031, Waterbury, Connecticut.



This eddy current testing device, shown inspecting thin-wall tube, detects flaws such as lap seams, embedded chips, cracks, tears and fractures.

ANACONDA®

COPPER TUBE

for Refrigeration and Air Conditioning

Anaconda American Brass Company

Elimination of costly reheat is the key to added controllability and flexibility in

Air Conditioning Interior Office Space

DAVID MICHAELI
Associate Member
ASHRAE

AVA TINFO

Various types of air conditioning systems have been designed for the interior spaces of modern office buildings. The systems are basically air, with some air-water combinations, with many variations as to pressure levels of air, degree of centralization of equipment, method and extent of control, etc. Most commonly used is the Central Rig, Single Duct, Reheat System, whose popular usage is dictated by economic considerations. With it, the cooled air is supplied to each floor or zone from a central point. Reheat is provided at each zone to match varying lighting, population and other interior loads.

Reheating, however, is inherently a costly process due to the waste of the cooling and heating energies involved. Consequently, the building engineer, in an attempt to reduce the operating costs, cools the supply air merely to the level required to satisfy the majority of the zones. As a result, comfort conditions are sacrificed on the more heavily loaded floors.

With such a system, any change in floor or zone loading (caused by increased lighting, additional office machines or a heavier occupancy) requires some make-shift provision such as packaged air conditioning units or rebalancing of the air system. The latter usually results in complaints from other floors or areas about the lack of ventilation or cooling capacity, and, at best, provides but a partial solution of the problem.

TO ELIMINATE REHEAT

With the basic principle in mind that costly reheat should be eliminated as a final control media (unless absolutely required due to humidity conditions), the writers have developed a system currently specified by them for two high rise office buildings,* to incorporate the flexibility feature and operating cost savings (achieved by eliminating both the sub-cooling and reheating) of a double duct system with the smaller space requirements and initial cost savings of a single duct reheat control system.

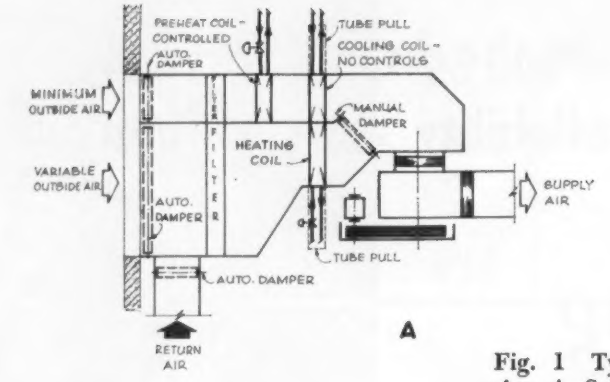
General arrangement—The central air supply rig, as shown in Fig. 1A, is split into two sections. The section handling minimum outside air is provided with an automatic minimum outside air damper, filters, preheat coils and cooling coils. The other section, handling all return air during the peak summer and winter cycles, and a mixture of return and outside air (or all outside air) during the intermediate cycle, is provided with automatic variable outside and return air dampers, filters and a manual balancing damper. Heating coils may also be provided at the designer's discretion.

For average design, with a total of one (1) cfm per sq ft of interior office area and one quarter (0.25) cfm per sq ft ventilation requirement, the minimum outside air section of the central rig is sized for 25% of the total air, with 75% being handled by the other section. Variation from this will not affect the general design principles of this system.

At each floor or zone, a secondary cooling-heating coil is provided and controlled from the space

David Michaeli and Ava Tinfo are Executive Engineers, Cosentini Associates, Consulting Engineers.

* 605 3rd Avenue, N.Y.C., office building—Fisher Brothers builders; Emery Roth and Sons, Architects. National Geographical Society Office Building, Washington, D.C.—Edward Durell Stone, Architect.



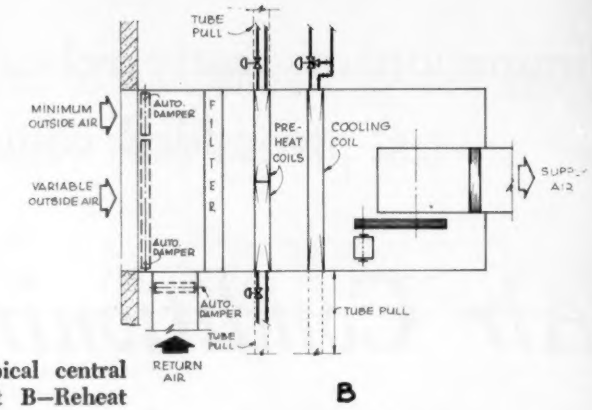


Fig. 1 Typical central rigs A–Split B–Reheat

itself. This coil is provided with 50 F secondary water during the summer cycle and hot water in the intermediate and winter cycle.

The secondary water system is sized for 4 gpm per 1000 cfm at an 8 F temperature rise in the summer cycle (50 – 58 F). Primary cooling coils are copper tube, copper fin, whereas secondary coils are copper tube, aluminum fin (since they do sensible cooling only, and remain dry during normal operating conditions).

Summer cycle operation–The minimum outside air section is designed to remove – during the summer cycle – all the latent heat load imposed on the system by both the ventilation air and the internal latent heat gain. With outside air at 95 F DB, 75 F WB as an example, an eight-row cooling coil with 42 F entering chilled water was selected to cool the air down to 50 F DB, 49 F WB. This air is mixed with return air at 75 F DB, 50% RH (selected room conditions), and the mixed air – at 69 F DB, 59.5 F WB – is delivered to the secondary cooling coil at each floor or zone. Assuming an average of 2 F fan and duct heat pick-up, a 6-row (600 fpm) secondary cooling coil, performing sensible cooling only, was selected to cool the air from 71 to 56 F. The system is hence capable of delivering air at any temperature between 56 F DB and 71 F DB as required by the space, while maintaining humidities down to 0.85 SHF. (See Psychrometric Chart – Fig. 4).

A heating coil provided in the return air section of the central rig will enable the building operator to supply some heat to the interior zones at start-up during the cooler weather of the summer cycle operation (when O.A. is 60 F or so). The geographical location of the building will determine the need for this heating coil.

Intermediate and winter cycle operation – The minimum outside air section is designed to heat the ventilation air from outside winter design conditions to 45 F. This 45 F minimum O.A. is mixed with an automatically controlled mixture of return and variable outside air to provide 54 F supply air to the secondary coils. The secondary coil, assuming two-heat pick-up, will heat the 56 F air to maintain selected space temperatures. (Hot water temperature should be maintained at a low level due to the deep coil provided for summer sensible cooling.)

Possible variations to system–Since two independent sources of energy are available at each zone, warm air and cold water in summer, cold air and hot water in winter, any number of the total number of zones–at the designer's discretion –can be converted easily to a double duct system. The hot and cold ducts change relative to the cycle of operation, the summer cold duct becomes the winter hot duct and vice versa. Duct sizes and controls must therefore be designed accordingly. Space thermostats for this system should be of the summer-winter type.

CONTROLS

Summer-winter change-over is manual. The change-over switch shall change the operation of the space thermostats from direct acting to reverse acting.

Winter cycle –

a. A duct thermostat sensing the temperature leaving the supply fan controls the variable outside air and return air damper, to maintain a fixed temperature of 54 F.
b. A duct thermostat, sensing the temperature leaving the minimum outside air preheat coil, controls a normally open, modulating valve in the steam line to the coil to maintain a minimum temperature of 45 F.
c. A low temperature protection thermostat in the supply fan suction shall stop the supply fan if the temperature entering falls below 40 F.

Summer cycle –

a. The minimum outside air and return air dampers open automatically when the supply fan is started. No other damper control is employed during the summer cycle.
b. The minimum outside air chilled water coil shall be uncontrolled.
c. A manual, open-close switch shall control a normally open valve on the mixed air heating coil, to provide for morning warm-up when necessary. A low limit thermostat sensing the temperature leaving the mixed air preheat coil shall over-call the action of the open-close switch to maintain a minimum temperature of 45 F leaving the coil.

Secondary coil – zone control – A summer-winter space thermostat shall control a normally closed modulating valve in the water line to the coil. The space thermostat shall be reverse acting for winter operation and shall be direct acting for summer operation. Summer-

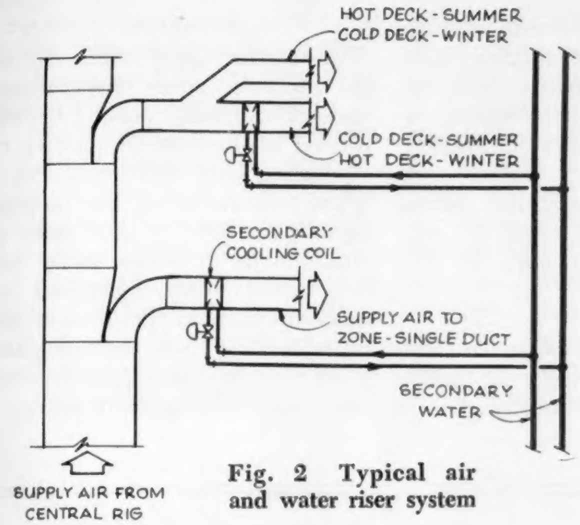


Fig. 2 Typical air and water riser system

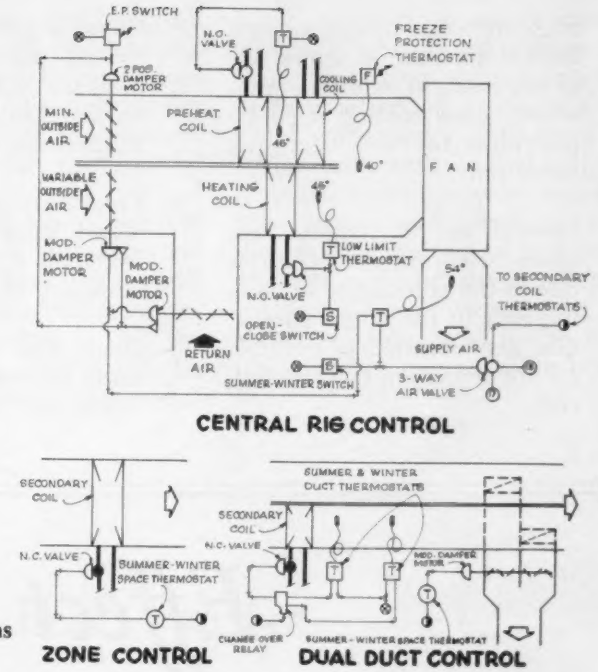


Fig. 3 Control diagrams

winter change-over shall be accomplished by the master summer-winter switch described previously.

Dual duct control –

a. Secondary coil – Summer and winter duct thermostats, sensing the temperature of the air leaving the coil, shall control a normally closed modulating valve in the water line to the coil. The duct thermostats shall be direct acting for summer operation and reverse acting for winter operation and shall maintain their set temperatures leaving the coil. Summer-winter change-over from one thermostat to the other shall be accomplished with a relay operated from the master summer-winter switch line.

b. Mixing dampers – A summer-winter space thermostat controls modulating dampers in the duct from the coil and in the by-passed duct, to provide supply mixed air. The space thermostat shall be direct acting during the summer time and reverse acting during the winter time. Summer-winter change-over shall be from the central master summer-winter switch previously described.

ADVANTAGES OF SYSTEM

1. Complete flexibility – loads handled at each zone from maximum to no load conditions. This is achieved with a central rig without space requirements of a double duct system.
2. Operating cost savings—waste of energy for sub-cooling and reheating eliminated.
3. Increased available refrigeration capacity as compared to a reheat control system since cooling is applied only where required and reheat is eliminated.
4. Increased cooling capacity of interior air system due to the availability of 56 F air at each zone, where and when required.
5. Possibility of converting part of the system, should same be desired, to a double duct system. Cold air always is available and its temperature is not affected by other zone requirements.
6. Savings in operating costs (re-

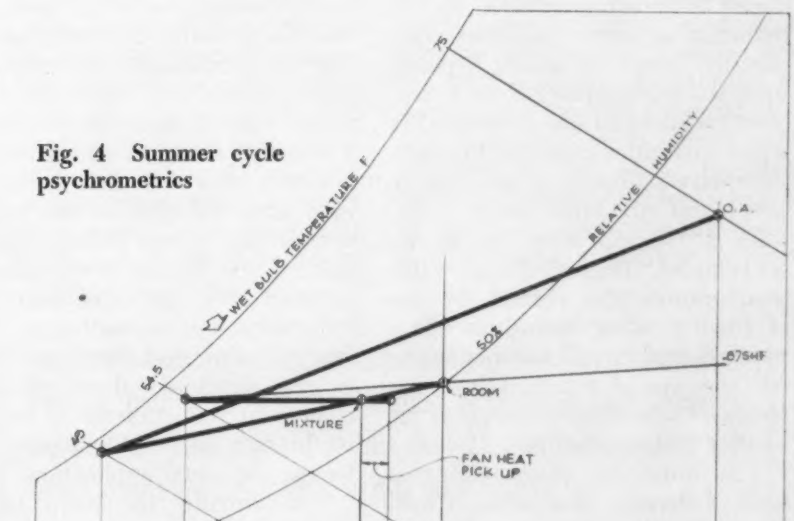


Fig. 4 Summer cycle psychrometrics

frigeration and reheat) when a limited number of zones are in use (such as overtime operation), since cooling need be provided for the occupied zones only.

7. Central air rig requires less space and simpler controls due to the elimination of cooling coils in the return air section and omission of controls for the minimum O.A. cooling coils.

8. Closer supply air temperature control due to smaller temperature difference between air and heating medium (low temperature water vs. steam).

9. Freeze-up possibility minimized since a separate preheat coil is provided for the minimum O.A.

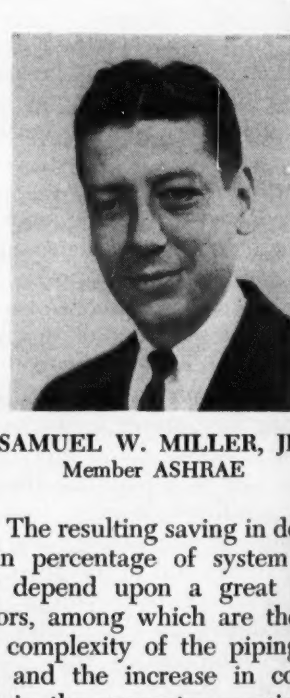
10. Steam trap maintenance normally encountered with steam reheat coils is eliminated.

SUMMARY

Without increasing either the cost or the shaft space requirements, and without relinquishing the central rig idea (while simplifying the central rig itself and decreasing its space requirements), the proposed method for interior office space air conditioning provides more flexible, more closely controlled, and more economical operation as compared to the reheat control system. With 71 F dry air supplied to each secondary coil, no over cooling at

---

Effect of Small Water



SAMUEL W. MILLER, JR.
Member ASHRAE

Principal economic factor underlying any hot water heating system is the water quantity circulated to handle the design heating load. The smaller the water quantity circulated, the more economical the distribution system will be. Stated another way, the higher the temperature drop of the water at design conditions, the more economical the system will be.

Generally speaking, both first cost and operating cost will be reduced in direct proportion to the reduction in water quantities. Specifically, however, much depends upon the relative costs of the equipment required to make the smaller water quantities possible. Primary-secondary pumping is one way in which the circulated water quantities and pump horsepowers can be reduced. Higher design water temperatures also permit the use of smaller water quantities. This paper describes still another method: the use of terminal heat exchangers specifically designed for smaller water quantities.

It must be recognized that each of these is, in a sense, a dual approach. Each contains elements which, if utilized correctly, can both reduce costs and improve performance. On the other hand, each also contains elements which can increase costs, and requires certain skills and knowledge on the part of the designer to avoid design and performance problems.

It is probable that the best design from a standpoint of both economy and performance must successfully integrate the potential advantages of all three.

Small water quantities reduce cost and improve control – The key to economy lies in the circulation of smaller water quantities. Reducing the design water quantity can result in substantial reductions in piping system cost, especially if the design base is the conventional 20 F drop. Halving the design water quantity (to a 40 F drop) theoretically cuts the size of the piping system and pump horsepower in half. While the cost is not actually reduced 50%, the attendant cost reductions are nevertheless substantial. For each 50% reduction in gpm circulated, there will be a coincident reduction of 20 to 30% in the first cost of the piping system in the usual application.

Eventually, the point of diminishing returns is reached. The greatest saving occurs from a 20 F drop to a 40 F drop. The next greatest from 40 to 80 F, and a smaller saving from 80 to 150 F. Little saving attends going from a 150 to a 200 F drop unless the system is quite large.

The resulting saving in dollars, or in percentage of system cost, will depend upon a great many factors, among which are the size and complexity of the piping system and the increase in cost, if any, in the apparatus required to utilize the smaller water quantities. Generally speaking, however, the cost of the distribution system (including pumps, piping, valves, fittings, specialties, pipe covering and control valves) will be a significant portion of the total system cost, and will offer the greatest avenue for potential savings. In conventional hot water systems for large buildings, the distribution system may comprise 10 to 20% or more of the total system cost, and in district heating systems even more. Significant reductions in overall cost are therefore possible as a result of reductions in the water quantity circulated.

Samuel W. Miller, Jr., is Chief Engineer, J. J. Nesbitt, Inc. This paper was presented as "Effect of Small Water Quantities on Heat Transfer Selection and Controllability."

a no-load condition is anticipated for this presently specified system, and hence no additional zone reheat coils are required.

The total static pressure of the system is increased by approximately 0.70 in W.G. This is due to an 8-row cooling coil in lieu of a normal 6-row coil at the central rig, and a 6-row dry coil at 600 fpm in lieu of a one-row reheat coil at each zone. The additional power consumption due to this increase in static pressure is rather insignificant considering all the advantages mentioned, and will be more than off-set by the savings in refrigeration and reheat energy normally wasted with a reheat control system of the conventional form.

It ought to be pointed out that the physical size of any individual zone will determine the overall degree of comfort obtainable. Individual room control is still the optimum system and should be used when economic considerations permit it or where special occupancy conditions require it. While the writers do not presume to have solved all problems associated with interior office space air conditioning, they do feel that the proposed system provides superior zone control and adaptability of the same central rig to individual room control for the small percentage of areas requiring it, as normally encountered in the interior spaces of modern office buildings.

Quantities on MTW heat transfer

The use of smaller water quantities for a given load also improves the quality of control performance in the terminal heat transfer apparatus. No system of heat distribution can be successful if it does no more than distribute the heat required for maximum load in the most economical fashion. The heat supplied must be capable of control at reduced capacities, and a high degree of control refinement may be necessary in process loads and many comfort applications. Where refined control is necessary, a control valve is almost invariably essential in some form. With valve control at any supply water temperature, the smaller the design water quantity, the better the control performance which can result. On the other hand, the higher the water temperature at a specific load and water quantity, the more difficult part load control may be to achieve.

Because smaller water quantities both reduce system cost and improve control performance, and higher water temperatures may increase system cost and adversely affect control performance, the assumption that the two go hand in hand is not necessarily valid. The designer must, essentially, work at cross purposes. He must successfully design a system which simultaneously utilizes the highest possible temperature drop and the lowest possible supply water temperature. One way this can be achieved is through terminal heat exchangers designed for small water quantities at lower temperatures.

Terminal heat exchangers designed for small water quantities — The supply water temperature and design temperature drop will have important effects on the selection and cost of the terminal heat transfer apparatus. The converse is also true, however: the selection of the terminal heat exchangers can have an important effect upon the water quantity circulated, and even upon the design water temperature which must be used. Just as the design water temperature is a controllable element in the design, so is the selection of terminal equipment for small water quantities. Where the choice is possible, the utilization of larger temperature drops in terminal equipment at lower temperatures may, in fact, result in more economy and better performance than the use of higher water temperatures.

Design temperature drops far larger than those customarily utilized in conventional design are possible today without increasing supply water temperatures. In many cases, standard equipment now on the market is suited for use with small water quantities at conventional supply temperatures, and is catalogued to permit easy selection on that basis. In other cases, equipment can be selected by the designer or readily modified by the manufacturer to meet

At the Medium Temperature Water Heating Symposium, held during the ASHRAE Semiannual Meeting in Chicago, four speakers tackled as many aspects of this subject. The other papers are being published separately in the JOURNAL.

April—Pipe and Pump Size Reduction, Homer Bird.

June—Effect of System Temperature on Pump Curve and Pressure-Drop Curve, G. F. Carlson.

July—Pressurization of MTW Systems, Raymond Harmon.

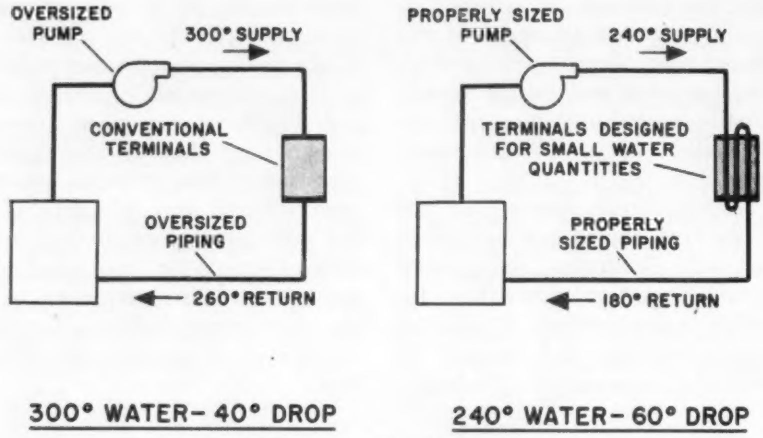


Fig. 1 Use of higher supply water temperatures does not necessarily result in smaller water quantities

the requirement. Temperature drops well in excess of 100 F are possible with LTW, and in excess of 150 F with MTW, in many classes of equipment. While not all heating equipment is susceptible to modification for use with smaller water quantities to the same degree, much opportunity remains for reducing costs and improving performance in low and medium temperature hot water systems through better utilization of terminal heat exchangers.

Terminal heat transfer equipment designed for use with large temperature drops at lower supply temperatures may not cost a great deal more than conventional equipment. In fact, there may be no cost increase at all. Even if the terminal equipment does increase in cost, however, the increase will usually be more than offset by the savings in the cost of the distribution system. In short, utilization of terminal heat exchangers designed for smaller water quantities at lower temperatures may permit substantial net savings in the cost of the system, and improve performance as well.

Higher water temperatures may not lead to economy – True economy in the system results from reduction in the circulated water quantity, not from higher temperatures. It cannot therefore be assumed that higher supply water temperatures automatically lead to smaller water quantities, less expensive heat exchangers and greater system economy. Smaller water quantities can be achieved in a number of ways, of which increasing the water temperature is but one. Heat exchangers are not inherently less expensive with higher water temperatures. The added cost of provision for operation at the higher temperature and pressure must be equated against the reduction in the amount of heating surface required. And finally, if the reduction in system water quantity can be achieved in some other way, the use of higher water temperature is not only unnecessary but undesirable.

Higher water temperatures inherently increase the cost of system components. The use of a higher temperature can therefore be justified only if it results in a saving in distribution system cost which more than offsets the added component cost. This may not always be the case in practice. The fact that control performance is adversely affected in direct proportion to increases in the supply water temperature adds another reason for caution.

This is not to say that higher supply water temperatures cannot lead to greater economy and better performance. They can, but only

Table I Comparison of the water quantities, main sizes, and pump horsepowers required to handle a 10,000,000-Btu/hr load at various design temperature drops

Design Temp. Drop	20	50	100	150	200
Gpm	1000	400	200	133	100
Main Size	10 in.	6 in.	4 in.	3½ in.	3 in.
Pump hp	15	7½	5	3	2

if the designer has already taken advantage of all the other possibilities open to him for reducing circulated water quantities, so that a further reduction can be achieved in no other way. This is especially true if the higher temperatures require the use of special materials or apparatus, or create unfamiliar or difficult design or operating problems.

Higher water temperatures may increase cost – Were all other factors equal, one could almost make an open and shut case that costs and problems go up directly as the supply water temperature is increased. As temperatures and pressures are increased, the costs of almost all system components – boilers, valves, fittings, water-to-air and water-to-water heat exchangers, expansion tanks, pumps – increase correspondingly. As temperatures increase, the problems of maintaining pressurization, of corrosion, of expansion, of achieving refined control, all increase proportionately. Above certain temperatures, extra equipment and specialized design techniques are necessary.

But for the fact that higher water temperatures can lead to smaller water quantities and less costly distribution and heat transfer equipment, higher water temperature systems would never even be considered. On the other hand, however, the assumption cannot be made that because higher water temperature systems do permit these economies that they are in themselves more economical. They may or may not be, depending upon the system and the skill of the designer.

The costs of system components do not increase uniformly with temperature, but rather, because of standardization of equipment into various temperature and pressure classifications, in steps. Not all system components increase in cost in the same steps, however, and the idea of what constitutes a more expensive system may depend upon the point of view. In general, hot water systems may be divided into three major temperature classifications: low, medium and high. Within each classification, changes in water temperature do not materially affect either cost

of system components or design techniques. Table II shows a comparison of selected system components required for the three classes of hot water systems. Note that some components are suitable for either low or medium temperature applications, without change, whereas others differ for all three classifications.

Generally speaking, it can be said that low temperature systems utilize standard equipment, that medium temperature systems utilize either standard or modified standard equipment at some increase in cost, and that high temperature systems utilize non-standard or special-purpose equipment, at a greater increase in cost. Moreover, the design and operating problems with medium temperature water are greater than those encountered with low temperature water, and with high temperature water still greater.

Higher water temperatures do not necessarily result in smaller water quantities – The assumptions often are made that higher water temperatures will result automatically in smaller system water quantities, and that higher design temperature drops are not possible without higher water temperatures. Neither of these assumptions is valid per se.

Higher water temperatures will result only in smaller water quantities if the designer has selected carefully all the terminal heat exchangers for their maximum ability to extract heat from the water circulated, and the components of the circulating system properly so they actually will perform as intended.

For example, many low temperature hot water systems are still designed for a 20 F drop. Most medium temperature water systems are designed for an 80 or 100 F drop. On the surface, it would appear that the use of the higher temperature results in a smaller water quantity. This may not always be the case. Let us assume a design using 300 F water and an 80 F drop. Should the designer, to be "conservative," oversize the heating equipment, piping, and control valves, to over-estimate the friction loss, and select a larger pump than necessary, the result is a system which actually operates at a 40 F drop and not 80 F. If, on the other hand, the designer had designed a 240 F system carefully, using heat exchangers utilizing temperature drops of 60 or 80 F or more, and sizing his distribution system and pump accurately, the result might well be a system which actually operates at a 60 F drop. In the latter case, the designer has been able to design a more economical system with a higher temperature drop and a lower supply water temperature. This example may seem extreme. Yet it is the author's opinion that many parallel cases can and do arise in practice, and that many designers are as yet a long way from taking full advantage of the possibilities for economy and better performance available at lower supply temperatures.

Since the use of higher supply temperatures will result in higher costs if a standard pressure classification is exceeded, and since the use of higher water temperatures may not necessarily result in reductions in circulated water quantities sufficient to offset the higher costs associated with the higher temperature, the designer must exercise considerable caution before making a decision to use increased water temperatures. For greatest economy, the alternatives of secondary pumping, and of terminal heat exchangers designed for smaller water quantities, should always be investigated before temperatures are increased into a higher pressure classification.

Smaller water quantities can result from better selection of heat exchangers – The water quantity which must be circulated to a heat exchanger to transfer a specific amount of heat is a function not only of the supply water temperature, but of the character and the entering and leaving temperature of the medium to be heated, and the design of the heat exchanger itself. In the generation of process steam or other high temperature process work, where the final temperatures required are close to the water temperatures, but little can be done to reduce water quantities by increasing temperature drop through the heat exchangers. On the other hand, where the temperature of the heated medium does not approach the supply water temperature, a great deal can be done. Comfort heating falls in the latter classification. In comfort

Table II Comparison of selected system components for various classes of hot water systems

System Class	Max Press. (1)	Max Temp. (F)	Heat Exchangers Water-to-air	Convector Radiation	Heat Exchangers Water-to-water	Boilers	Control Valves
Low	30	250	Copper Tubes and Headers	Copper, Steel or Cast Iron	Standard Low Pressure	Low Press. Heating Boilers Cast Iron or Steel	Standard for Most Manufacturers
Medium	100	325	Red Brass Tubes Steel Headers	Steel only direct. Copper may be used with diverting valves and secondary pumps		High Press. Steel only Fire Tube (2) or Water Tube	
High	250	400	Cupro-Nickel or Admiralty Metal Tubes Steel Headers	Steel only direct. Copper requires heat exchanger	High Pressure (Special)	High Press. Water Tube	Cast Steel Stainless Trim

(1) to prevent boiling (psi) (2) fire tube boilers increase in cost by pressure steps

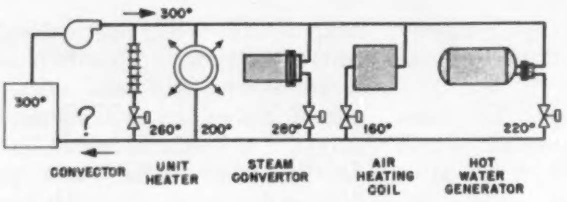


Fig. 2 The concept of "overall design temperature drop" is not applicable to large hot water systems because the capability of equipment to utilize large temperature drops varies widely

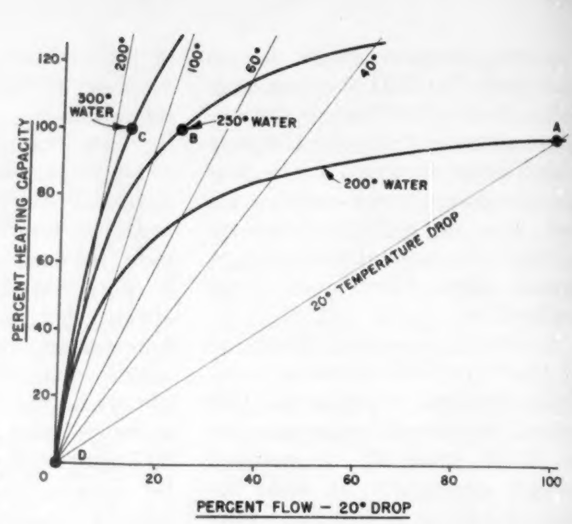


Fig. 3 Capacity-flow characteristics of a typical hot water heating element

applications, final air temperatures above 135 F are rarely desirable, and lower temperatures often may be required. Domestic water heating rarely requires temperatures above 180 F. In both these applications, which together may comprise a substantial portion of the total heating load, there is considerable area for the reduction of circulated water quantities without increasing design water temperatures.

Standard heating equipment is susceptible to modification or adaption for use with smaller water quantities in varying degrees, depending upon its type. Convector radiation in its present form is virtually unsuited to use with smaller water quantities. Unit heaters and water-to-water heat exchangers are better suited. Heat transfer surface used for air heating in fan systems, which may either be designed by the manufacturer or selected by the engineer to meet specific requirements, is best suited. In many cases, design temperature drops far higher than those customarily used in most design today can be utilized with the equipment of this type. Table III shows a comparison of the temperature drops actually possible in various types of commercially available heating equipment, with the temperature drops used in customary design practice.

It should be noted that the temperature drops listed are from standard catalogued ratings for equipment items currently available. It is probable that even higher temperature drops will be possible in future equipment, especially in the field of convector radiation, as the use of higher temperature water becomes more common and as more manufacturers begin to design equipment especially for smaller water quantities.

The assumptions are frequently made that return water temperatures much below 200 F are either undesirable or cannot be achieved with economical heat exchangers. Neither is necessarily true. The problem is not one of returning water to the boiler at too low a temperature. Outdoor reset controlled heating systems regularly return water to the boilers at temperatures as low as 100 F, and even lower in dual temperature systems. Nor is it one of being able to extract the heat. There is still a great deal of heating capacity remaining in 200 F water (as evidenced by the large number of systems which are designed with 200 F as a supply temperature) and there is no reason why this heat should not be utilized in systems using higher supply temperatures, if it can be done so economically. There is ample evidence that it can be.

Heat exchangers for small water quantities may not increase cost – Rarely will the increase in cost of the heat transfer surface to make smaller water quantities possible be greater than the resulting savings in the cost of the piping system. In many cases, there is little or no increase in cost of the heat exchanger at all as a result of its adaptation for use with a smaller water quantity, if the surface has been properly designed and circuited for the application.

The quantity of heat transferred from a water-to-air heat exchanger is expressed by the formula:

$$H = U \times A \times MTD$$

U = Overall Coefficient of Heat Transfer
A = Air-Side Surface Area
MTD = Log Mean Temperature Difference—Air-to-Water

Most heat transfer surface for conventional hot water application utilized a number of parallel water flow paths. In a typical standard heat exchanger designed for a 20 F drop, reducing the design water quantity reduces both the water velocity (which lowers U) and the MTD, reducing the capacity. For small water quantity application, however, this surface can be recircuited to reduce the number of flow paths. Recircuiting, if properly done, increases the water velocity, which in turn increases U, often by a percentage approaching that of the percentage reduction in MTD, depending upon the initial selection. Any small difference can be made up, if necessary, by increases in the airside surface area A. Hence the same surface essentially produces the same capacity with a far smaller water quantity, if properly recir-

LONG RANGE PLANNING COMMITTEE CHARTS AIMS AND ACTIVITIES OF SOCIETY FOR THE FUTURE PAGE 56

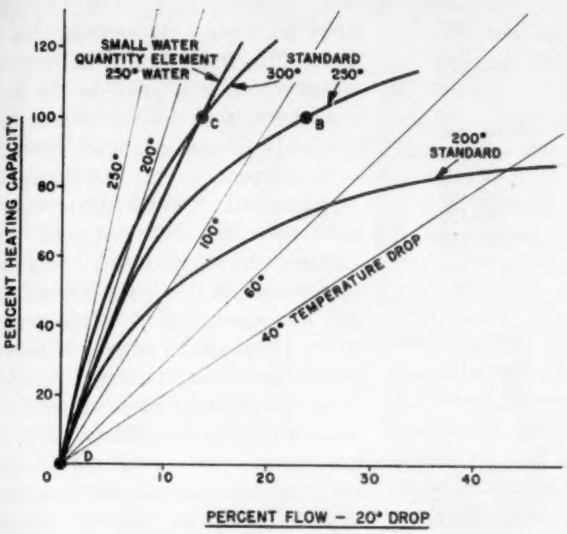


Fig. 4 Capacity-flow characteristics of standard heating elements and elements designed for small water quantities

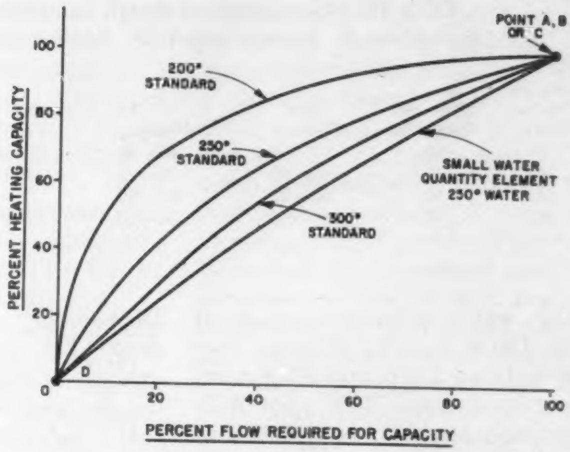


Fig. 5 Comparison of capacity-flow characteristics where flow is expressed as a percent of full load flow

cuited. The resulting increase in cost of the surface, if any, is negligible.

Many manufacturers do not discriminate at all in price between heat exchangers designed and circuited for small water quantities, and those designed for conventional application, so long as the standard pressure classification is not exceeded, or a great amount of additional surface is not required. It is, therefore, often possible, simply by proper selection of the heat exchangers, to reduce the water quantity circulated, and hence the piping system cost, without increasing the cost of the terminal units at all.

Even if the cost of the terminal heat exchange elements were increased as a result of the need for additional surface, however, such increases in cost will usually be small compared to the possible savings in the circulating system. (Note that this is the heating element cost only, not the cost of the total terminal equipment.) It is of special interest to note in this connection that the cost of the additional surface required to use smaller water quantities at the same water temperature is in many cases no more than the cost of the construction required if higher temperatures are used.

Determining the most economical design – The determination of the most economical supply water temperature requires a careful analysis of many factors. The designer must weigh the possible economies associated with the smaller water quantities and the smaller heat exchangers against the more costly equipment required to withstand the higher temperatures and pressures, and against the design and operating problems associated with the higher temperatures. This evaluation must be made for the specific system in question.

While no rule of thumb can be used in all cases, it is the author's opinion that the following approach can lead to the most economical design:

1. **Utilize the highest water temperature consistent with good practice within the class of system being considered, but in no case higher than necessary for minimum water quantities or to meet process requirements.**
2. **Design each of the terminal heat transfer units for the smallest water quantity which will provide the required heating capacity, using equipment specifically selected for the purpose. If necessary and possible, increase the amount of surface in the terminals to obtain the high temperature drops desired.**
3. **Design the distribution system carefully, using precise data and a minimum of so-called "safety" factors in pipe and control valve sizing and pump selection. Take advantage of secondary pumping where possible.**

This approach assures the use of the highest water temperature which can be used in a system whose components are of a given cost, and at the same time assures that every opportunity has been utilized to reduce circulated water quantities by means of more efficient terminal heat exchange devices and secondary pumping.

In practice, this approach would lead to water temperatures around 240 F for low temperature systems, about 300 to 325 F (depending upon the type of pressurization) in medium temperature systems, and water temperatures around 400 F in high temperature systems. It would lead to temperature drops in many types of equipment 60 to 100% greater than those now customarily used for design. It would lead to the design of systems which provide the economies of small water quantities without the costs of higher supply temperatures. In many cases, it would make LTW systems more economical than MTW systems, and MTW systems more economical than HTW systems.

The greatest economy results from the use of the highest water temperature and the smallest water quantity within a specific class of equipment requirements. Thus,

Table III Comparison of design temperature drops possible in various types of heating equipment

Supply Water Temp.	Convector Radiation	Unit Heaters	Fan Apparatus Heating	Fan Apparatus Ventilating	Temp. drop in common use
180	20	30	40– 80	60–100	20
240	30	40– 80	80–120	100–130	20– 60
320	40	60–120	120–160	140–180	80–100
400	40	150–200	180–220	200–240	150–200

240 F water is more economical than 180 F, and 325 F more economical than 270 F, and 400 F more economical than 375 F. But it is also probable that, except in unusual cases, 240 F water is more economical than 270 F, and 325 F more economical than 375 F. This is especially true if the designer of the system in the lower pressure classification avails himself of the opportunities to reduce system water quantities by the use of more suitable terminal heat exchangers.

It is interesting to note in this connection that most of the economies claimed for HTW systems are really economies which result from the use of small water quantities (high temperature drops) or from the use of hot water per se. Most of the comparisons made to illustrate the economy of HTW are made either with high pressure steam or with low temperature water systems designed for the conventional 20 F drop. A valid cost comparison between a HTW system, a MTW system designed for a 150 F drop, and a LTW system designed for a 100 F drop might be most revealing in this regard.

The author recognizes that this option is not always available. Process loads, especially steam, may dictate higher water temperatures. Or heating equipment of a type not readily adapted for selection for larger temperature drops may predominate. For extremely large district distribution systems, certainly a strong case can be made for the use of HTW. But for local distribution within a building, where a significant portion of the heat transfer apparatus is of the type which lends itself to selection for smaller water quantities, consideration should always be given to utilizing terminal heat exchangers designed for small water quantities at supply temperatures low enough to minimize the design and operating problems.

Determining design temperature drop – Let us assume that after an analysis of all the factors involved, the designer has chosen to utilize a MTW system, and has set about to design the piping system and to select the terminal heat transfer equipment. He still is confronted with a number of decisions, and his decisions will largely determine the economy and performance of the system when it is installed and operating. His first decision lies in determining the water flow required for the terminal heat exchangers, and here he comes against his first stumbling block.

The use of MTW instead of LTW will usually permit either larger temperature drops in the heat transfer equipment, or reduced size, less expensive (except for pressure construction) heat transfer surface. Because of the higher available mean temperature difference between the water and the load (entering air in water-to-air devices), the designer is usually able to exercise a choice between reducing the size of the piping system or reducing the size and cost of the terminal heat transfer elements.

For reasons previously stated, the best approach usually lies in designing the system for the highest possible temperature drop. But recognition must be made of the fact that terminal heat exchangers vary widely in their ability to extract the necessary heat at a given temperature drop, and the question arises: "How does the designer determine the design temperature drop?"

Concept of overall temperature drop handicaps economical design – The answer is that the system, for greatest economy, should not be designed on an overall temperature drop. It should be designed on the basis of the minimum flow in gpm or lb per hr required to handle the load at each terminal heat exchange element in the system. This flow should be the minimum which will provide the necessary heat if the heat exchanger is properly designed and circuited, not necessarily the minimum flow to standard devices designed and cataloged for conventional applications. In other words, each heat exchanger in the system is selected for the maximum possible temperature drop, and this requirement is converted immediately to flow so that the piping system may be designed on the flow basis. With this approach, the smallest possible total water quantity automatically results, and the overall temperature drop in the system becomes of no consequence to the designer.

This is as it should be. Design temperature drop does not exist at all except in the mind of the designer. The probability of design temperature drop occurring in actual practice is so remote as to be classified as a freak accident, especially in comfort systems utilizing outdoor reset control. In order for design temperature drop to occur in operation, all loads, friction losses, pump and equipment capacities, and flows would have to occur simultaneously exactly as assumed for the design – an obviously remote combination of possibilities. Moreover, temperature drop has no meaning (other than as it might change the viscosity) in the circulating system. The system circulates a fluid which conveys heat, not heat itself, and cannot discriminate between a volume of fluid from which a great deal of heat is to be extracted at the terminal and one from which there will be none. In addition, as soon as any part of the system comes under control, as it will do, the flows and temperature drops throughout will be quite different from anything assumed by the designer.

Temperature drop must be recognized for what it is: a design tool for determining the fluid flow to a heat exchanger at design conditions, and nothing more. It is a tool handed down to us from previous hot water practice in the residential field where the circulated water quantities meant little or nothing in the total system cost, and where the character of the terminal equipment was largely unsuited to the use of small water quantities in any event. Properly

used, temperature drop can be a very good tool, if any tool is needed, which it may not be. Improperly used, it can inhibit economical design and cause problems.

Using the same overall temperature drop to establish the design water quantities for the heat exchangers in a large hot water system constitutes an improper use of temperature drop, since it ignores the varying capabilities of various types of equipment. Using temperature drop to size piping is even worse. Both approaches will lead to unnecessarily expensive systems, or operating problems, or both.

Equipment capabilities vary — Standard comfort heating equipment designed for low temperature water generally requires large water quantities for satisfactory performance, even if high supply temperatures are used. For this reason, equipment used should be designed and circuited expressly for small water quantity application.

For example, a short length of fin-tube radiation is impractical to use with 300 F water and a 150 F drop. While it is theoretically (and under carefully controlled conditions actually) possible to obtain such a drop, in practice the water quantity would be so small as to create problems of flow regulation, air binding and stratification of both water within the tube and final temperature along the length. For this reason, fin-tube radiation usually is limited to temperature drops of 40 to 60 F even in long lengths, regardless of supply temperature. (Note: It is with this type of equipment that primary-secondary pumping can be used to greatest advantage.) Standardized fan units are better suited to large temperature drops. Properly selected and designed propeller fan unit heaters with 300 F water can be used with design temperature drops on the order of 100 F or more.

Heat transfer elements in fan systems which have been designed by the manufacturer or can be selected by the engineer for the specific application, however, can readily utilize temperature drops as high as 150 F or more with 300 F water. Hot water (and to a lesser degree, steam) convertors can also often be designed for smaller water quantities. Process equipment of various kinds might have similarly varying capabilities depending upon the equipment provided and its suitability for use with small water quantities.

For this reason, unless all heating and process equipment on a job is identical in kind or capability (which is usually not the case), the designer must recognize the varying capabilities for use with small water quantities inherent in the equipment used in the system; otherwise, he has either unnecessarily increased the size and cost of the distribution system or run the risk of some equipment not heating properly.

Determining design flows—Recognition of the varying capabilities in the design can be achieved by arbitrarily assigning a temperature drop to each class of equipment. A better way, and the one which leads to the greatest economy, is to select each item of heat transfer apparatus for the smallest water quantity which produces the required heating capacity.

Temperature drop may properly be used to determine flow for each specific item of equipment if no other information is available. In many cases, however, manufacturers of equipment designed specifically for use with small water quantities list capacities varying directly with flow in gpm or lb per hr, making the use of the temperature drop step unnecessary. In either case, the most economical piping system will result if the smallest quantity which will provide the required heating capacity is selected. Caution should be exercised with either approach to insure that the equipment is selected for its maximum capability. In other words, the water quantity selected must be adequate to provide the required heating capacity, but not so large as to result in capacities greatly in excess of the capacity desired. It must be remembered that any specific hot water heating device does not have a fixed maximum capacity, but rather a maximum capacity which varies with maximum flow. Too much flow not only increases the cost of the circulating system but increases control problems as well. Regardless of the method used for determining flow in the heating equipment, the piping and pumps are selected on the basis of flow.

The importance of this discrete approach, and the economy and improved performance which can be achieved by its use, will vary with system size and the type of terminal heat exchangers utilized. The larger the distribution system, and the greater the proportion of terminal units which are subjective to selection for small water quantities, the greater will be the economy. The use of this approach is possible at any design water temperature. Since economy lies in small water quantities rather than in higher temperatures, its use (accompanied by the judicious selection of heat exchangers) will often permit the design of a system which is not only the most economical possible at a given supply temperature, but which also may be more economical than a system utilizing higher temperatures but designed by conventional means.

(Note: Knowledge of the design temperature drop through each item of heating equipment is desirable as an aid to system balancing. But, it should be determined after the capacity and flow have been established. The resulting temperature drop in each equipment item should be computed or read from catalog data and listed in the equipment schedule as a guide to the installing contractor in balancing the system. Knowledge of the approximate return water temperature may also be desirable in determining design pressure drop.)

Effect of small water quantities on controllability—Essentially, the influence of design water quantity on controllability can be summarized thus: for a specific load and water temperature, the higher the design temperature drop, the better will be the control performance at reduced loads. Higher supply temperatures do not automatically lead

1960 TRANSACTIONS, FOLLOWING CLASSIC STYLE OF PRECEDENT VOLUMES, NOW AVAILABLE PAGE 76

1. Use of higher design temperature drops (smaller circulated water quantities) in a hot water system reduces first cost and operating cost and improves control performance.

2. Smaller water quantities may be achieved by means of higher water temperatures, primary-secondary pumping, or by the use of more efficient terminal heat exchangers, or by all three in combination.

3. With properly designed terminal heat exchangers, much smaller water quantities than are now being used in customary design practice are possible without increasing supply temperatures.

4. Increases in supply water temperature increase the cost of system components. Costs increase in steps in accordance with standard temperature and pressure classifications.

5. Properly designed terminal heat exchangers can result in smaller system water quantities at a cost which is often lower than that resulting if water temperatures are increased into a higher classification.

6. The concept of design temperature drop, while useful in establishing design flows in individual terminal heat exchangers, may inhibit economical design if applied to the system as a whole, since individual heat exchangers may vary widely in their ability to provide the required heating capacity at a given temperature drop. Temperature drop should never be used for pipe sizing.

7. The most economical system design at any temperature will result from a discrete approach in which the piping system is designed on the basis of flow and each heat exchanger is selected to provide the heat required with the smallest possible flow. Terminal equipment used must be especially designed and circuited for small water quantity application.

8. The most economical and satisfactory system of all can result from the careful integration of all the advantages of higher temperatures, primary-secondary pumping and more efficient terminal heat exchangers.

9. It is probable that the proper use of these three principles in general practice would do as much to reduce costs and improve performance in hot water heating systems as any single development the industry has yet achieved.

10. There is a need in the industry for more adequate design information which can lead to the practical everyday use of the basic economies and performance advantages possible in hot water heating systems.

to better control, however, because, conversely, for a specific load and temperature drop, the higher the design water temperature, the more difficult refined control at part load will become.

Characteristics of water-to-air heat exchangers — An understanding of the fundamental capacity-flow characteristics of a hot water heat exchanger is essential to an understanding of the influence of small water quantities on controllability. Fig. 3 shows the capacity flow characteristics of a typical hot water heating element for air heating. Per cent heating capacity is plotted against flow for several entering temperatures, with lines of temperature drop superimposed.

As flow is throttled in a heating element designed for 200F water and a 20 F drop, little reduction in capacity takes place until temperature drops of 60 to 80 F are reached, after which the capacity drops off more rapidly. In other words, significant capacity reductions do not occur until flow has been reduced to less than 30% of design flow. In order to provide refined control in the range of 0 to 20% of capacity, the valve must be capable of refined control between 0 and 5% of design flow. There is no commercially available valve that can meet this requirement.

With higher water temperatures and the same heat exchanger, the capacity requirement is met with progressively higher design temperature drops, and 100% flow now occurs at a higher temperature drop, as at point B or C rather than A. Note how the circulated water quantities have been reduced drastically for the same capacity requirement by the use of higher water temperatures. Note also that with respect to the maximum flow necessary to meet the capacity requirement, curve CD is more nearly linear than curve BD, and BD more nearly linear than AD.

If the heat exchanger were recircuited (and possibly surface added) to obtain the same capacity with 250F water as was previously available with 200F water at the same flow, another improvement in performance is made. This curve is shown in Fig. 4 superimposed on the data from Fig. 3. Note that the characteristic here is even more linear than that for 300F water with a standard element. Note also that the modification of the heat exchanger provides the same temperature drop, and hence the same economy in the distribution system, without the possible costs and problems associated with the higher water temperature.

Fig. 5 shows the same data plotted in a different way. Instead of plotting per cent capacity against per cent flow for 200 F water and a 20 F drop, per cent capacity is plotted against the percentage of flow required to obtain the capacity desired. Note that as the design water temperature drop is increased, the characteristic becomes more nearly linear for a standard heat exchanger, but that the most nearly linear characteristic of all results from the terminal heat exchanger designed to provide the higher temperature drop at the lower supply temperature.

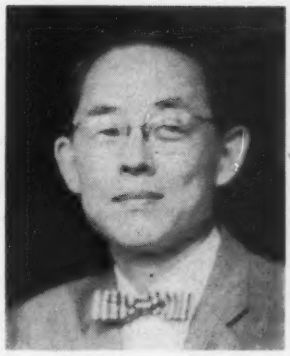
The controllability of any hot water heat exchanges is determined by the rate of change of capacity with changes in flow, and the more nearly linear the capacity flow curve becomes, the more easily controllable it will be. With the more nearly linear curve, the valve is wider open at any reduced capacity, making good control possible with valves which are commercially available. Higher design temperature drops (smaller water quantities for a given load) therefore result in improved controllability. The greatest improvement in controllability will result if the smaller water quantity is achieved at lower supply temperatures by better utilization of terminal heat exchangers.

The use of higher design temperature drops, even at lower supply temperatures, although it greatly improves control, will not inherently insure good control performance. Other considerations, such as properly characterized valves, and a pressure drop through the valve at design flow which is

(*Continued on page* 100)

Graphical analysis of a

Cross-flow Cooling Tower



HIDEO UCHIDA
Member ASHRAE

Analyses of cross-flow cooling towers have been provided previously by several authors. Snyder's theoretical analysis, assuming a linear relationship between water temperature and the corresponding enthalpy of saturated air, resembles that of cross-flow heat exchangers, except that enthalpy potential is used for cooling towers, whereas temperature is used for heat exchangers. It was Snyder's purpose to obtain experimental results. Zivi and Brand use the same differential equation presented in this paper, giving the water temperature distribution in the fill of a cross-flow cooling tower. This procedure is suited for automatic computation under the trial and error method.

GRAPHICAL ANALYSIS

Fig. 1 is a schematic vertical section through the direction of the air flow. Water entering the top of the tower flows uniformly and at the same temperature down through the fill. Direction of the air flow is defined as positive x-direction while the positive y-direction is the direction of water flow. Total length of the air flow and water flow are X and Y, respectively. Air conditions, temperature and enthalpy are the same along the entire length Y. Through any z-direction which is perpendicular to both x and y, the water temperature and the air enthalpy are assumed to be the same. Where the width of the tower is Z, the analysis of cross-flow cooling tower can be treated as a part of a two-dimensional problem.

Incremental element of volume is shown in Fig. 1. Water temperatures entering and leaving the element of volume are t_{wxy} and $t_{wx,y+\Delta y}$, respectively. Enthalpies of saturated air, the thin air films over the surface of water entering and leaving the element of volume are i'_{wxy} and $i'_{wx,y+\Delta y}$, respectively. Air enthalpies entering and leaving the element of volume are i_{xy} and $i_{x+\Delta x,y}$.

Heat quantity transferred between air and water in the incremental element of volume, the symbol ΔQ given the following relationships:

$$\Delta Q = \frac{L}{X} \Delta x \,(t_{wxy} - t_{wx,y+\Delta y}) = -\frac{L}{X} \Delta x \,\Delta t_{wxy} \qquad (1)$$

Hideo Uchida is a professor at the University of Tokyo, Japan. This paper was presented at the ASHRAE Semiannual Meeting, Chicago, Ill., February 13-16, 1961.

Fig. 1 Vertical section through direction of air flow

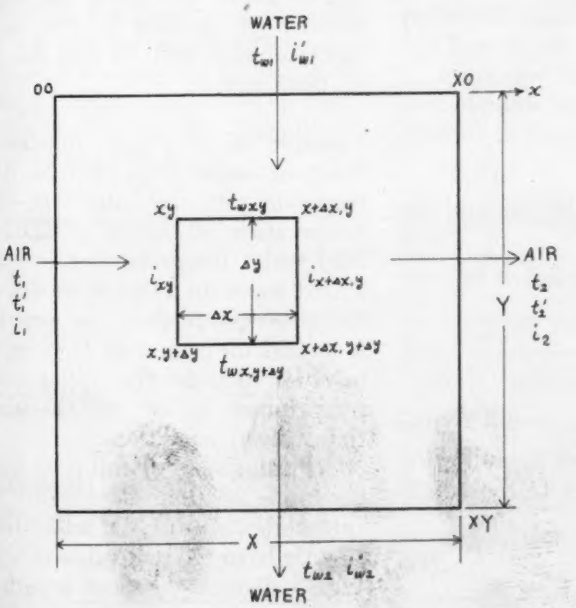
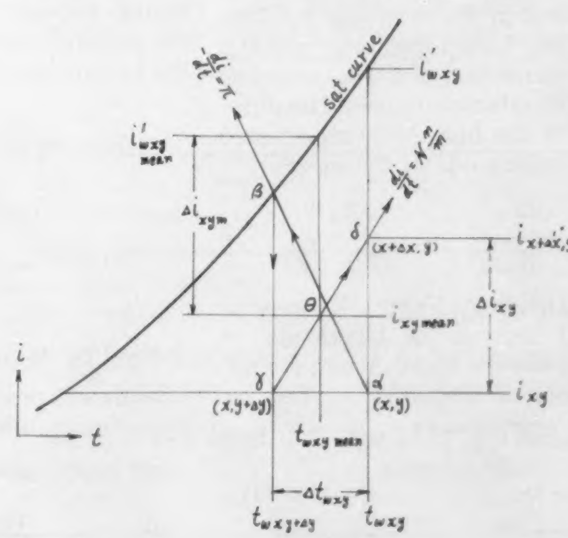


Fig. 2 Part of the t-i diagram of humid air



$$= \frac{G}{Y} \Delta y \, (i_{x+\Delta y, y} - i_{xy}) =$$

$$\frac{G}{Y} \Delta y \, \Delta i_{xy} \qquad (2)$$

$$= KaZ \, (i'_w - i)_{mean} \, \Delta\chi \, \Delta y = KaZ \, \Delta\chi \, \Delta y \, \Delta i_{xym} \qquad (3)$$

$$\doteqdot \tfrac{1}{2} KaZ \, \Delta\chi \, \Delta y \, [(i'_{wxy} + i'_{wx, y+\Delta y}) - (i_{xy} + i_{x+\Delta x, y})] \qquad (4)$$

Assuming constant overall heat transfer coefficient Ka in the tower, the transfer unit U of a cross-flow cooling tower can be represented thus:

$$U = \frac{KaV}{G} = \frac{1}{Y} \sum \frac{\Delta i_{xy}}{\Delta i_{xym}} \Delta y \qquad (5)$$

$$\frac{U}{N} = \frac{KaV}{L} = \frac{1}{X} \sum \frac{-\Delta t_{wxy}}{\Delta i_{xym}} \Delta\chi \qquad (6)$$

Using integral forms, those relations are

$$U = \frac{1}{Y} \iint \frac{\frac{\partial i}{\partial \chi}}{i'_w - i} \, d\chi \, dy \qquad (7)$$

$$\frac{U}{N} = \frac{1}{X} \iint \frac{-\frac{\partial t_w}{\partial y}}{i'_w - i} \, d\chi \, dy \qquad (8)$$

Fig. 2 represents a part of the t-i diagram of humid air. Intersection of the t_{wxy}-line and the i_{xy}-line is point α (x,y). Point γ is the intersection between the $t_{wxy+\Delta y}$-line and the i_{xy}-line, Point δ the intersection between the t_{wxy}-line and the $i_{x+\Delta x, y}$-line, and point β the intersection of the $t_{wx, y+\Delta y}$-line and the saturation curve. The direction of the line γ-δ is represented by dividing eq. (1) by eq. (2).

$$\frac{di}{dt} = \frac{\Delta i_{xy}}{-\Delta t_{wxy}} = \frac{L}{G} \frac{\Delta\chi}{X} \frac{Y}{\Delta y} \qquad (9)$$

Dividing length X into m equal sections of length $\Delta x = X/m$, and the length Y into n equal sections of length $\Delta y = Y/n$, eq. (9) is expressed as

$$\frac{di}{dt} = N \frac{n}{m} \qquad (10)$$

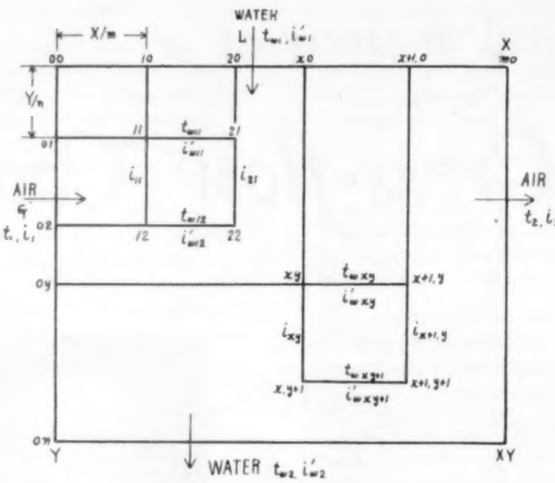


Fig. 3 Division of flow length of water and air into six sections

Therefore, when the temperature drop of water through the incremental element of volume is known $\Delta t_{wxy} = [t_{wxy+\Delta y} - t_{wxy}]$ Δt_{wxy} will be found negative, and with the given water-air ration N, point δ can be obtained on the line γ-δ with a direction. $\frac{di}{dt} = N \frac{n}{m}$.
Then the water temperature $t_{wx, y+\Delta y}$ and the air enthalpy $i_{x+\Delta x, y}$ can be found, and the element of volume remains.

Average conditions of air and water in the incremental element of volume are point θ in Fig. 2, the middle point of the line γ-δ. Reading the values of the point θ, it is possible to get the average enthalpy of air i_{xym}, the average temperature of water t_{wxym}, and the average enthalpy of saturated air over the water i'_{wxym}. Enthalpy difference between the air and saturated air over the water, Δi_{xym}, the enthalpy potential difference in the incremental element of volume.

We refer π to the $\frac{di'}{dt}$ of the direction of the line α-β as follows:

$$\pi = \frac{\beta\gamma}{\gamma\alpha} = \frac{i'_{wx, y+\Delta y} - i_{xy}}{t_{wxy} - t_{wx, y+\Delta y}} \qquad (11)$$

Putting $\beta\gamma \doteqdot \Delta i_{xym}$, in differential triangle $\alpha\gamma\delta$ produces the following approximate relation to the eq. (6)

$$\frac{di'}{dt} = \pi \doteqdot \frac{mN}{U} = \frac{mL}{KaV} \qquad (12)$$

As seen in eq. (12), π is an approximate definite value in the tower. Therefore, if we get point β, which is the intersection between the saturation curve of t-i diagram and the line of the direction $-\pi$ drawn from the point α, we can get the point γ from the point β and the point δ from the point γ. Then, it is possible to have the water temperature and the air enthalpy entering the next incremental element of volume. When this procedure is continued, alternately, as shown in Fig. 4, the distribution of water temperatures and air enthalpies in the tower can be determined.

Substituting those values obtained through the procedure into eq. (5) and (6), approximate values of U, U/N and Ka, respectively, can be found, and using the larger number of m and n, the more exact solution of U/N and Ka can be determined.

Example of graphical solution – Inlet air enthalpy $i_1 = 38.54$ (corresponding to the inlet wet-bulb temperature of air $t_1' = 82.04$ F), inlet water temperature $t_{w1} = 98.6$ F and water-air ratio N = 1.2 are the given conditions. The problem is to find the values of U/N or Ka in order to make the outlet water temperature $t_{w2} = 87.0$ F under those given conditions.

Putting m = 6 and n = 6, assuming $\pi = 6/1.83 = 3.28$. The selected value 1.83 will differ slightly from the real value of U/N. Fig. 3 divides the flow length of

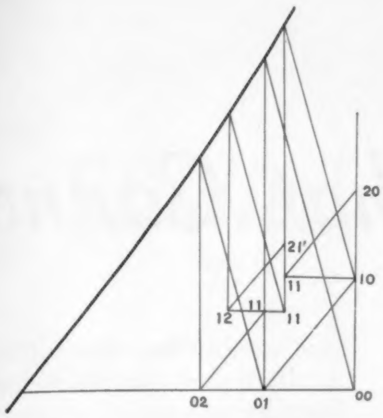


Fig. 4 Part of the graphical analysis of the tower

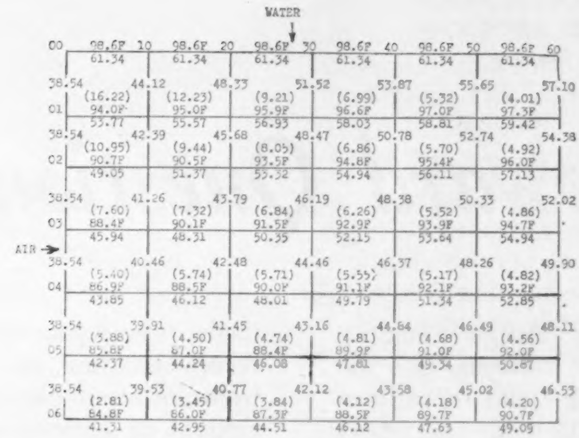
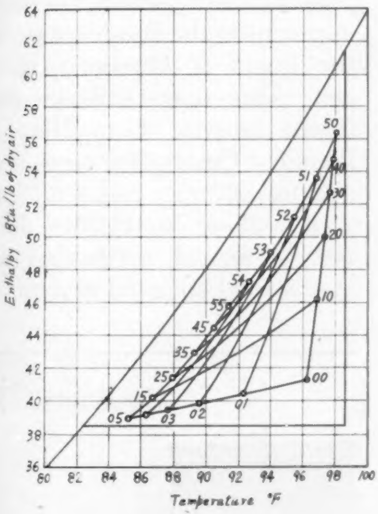


Fig. 5 Final results of analysis

Fig. 6 Average conditions of water and air in tower



water and air into 6 sections. Fig. 4 shows a part of the graphic analysis of the tower; line $\alpha\beta$ with di/dt $= -3.28$, and the line $\gamma\delta$ with di/dt $= 1.2$ can be drawn by the same procedure presented in Fig. 2. The final result is represented in Fig. 5. From the figure, for example, the temperatures and enthalpies for the incremental element of volume figured by 22-23-33-32 can be determined.

$i_{23} = 43.79$, $i_{39} = 46.19$, $t_{w22} = 93.5$, $t_{w23} = 91.5$, $i_w'{}_{22} = 53.32$, $i'_{w23} = 50.35$, $\Delta i_{22m} = 6.84$, $\Delta i_{22} = 46.19 - 43.79 = 2.40$, $\Delta t_{w22} = 93.5 - 91.5 = 2.0$

From this the mean temperature of outlet water $t_{w2} = 87.0$ F and the outlet enthalpy of air $i_2 = 51.3$ can be determined, and it is evident that the initial value of $\pi = 3.28$ was good, for it resulted in $t_{w2} = 87.0$ F.

Substituting those values shown in Fig. 5 into eq. (6), the U/N can be calculated as 1.765. The real value of U/N = 1.765 is the difference of 3.7% from the value as 1.83 given in π. If we select any value for V/L, the value of Ka, which represents the performance of the tower, can be obtained, substituting U/N = 1.765 into eq. (6).

Fig. 6 shows the average conditions of water and air in the tower.

NOMENCLATURE

Y = Length of air flow perpendicular to water flow
X = Length of water flow perpendicular to air flow
Z = Width of tower perpendicular to X and Y
V = Total volume of tower
t = Dry bulb temperature of air, F
t' = Wet bulb temperature of air, F
i = Enthalpy of air, Btu/lb of dry air
t_w = Temperature of water, F
i'_w = Enthalpy of saturated air temperature of which is t_w, Btu/lb of dry air
L = Quantity of water flow, lb/hr
G = Quantity of air flow, lb of dry air in humid air per hr
K = Overall heat transfer coefficient, Btu/ft²hΔi
a = Ratio of contacting area between air and water, unit area per unit volume of tower
N = L/G = Water air ratio
U = Number of transfer unit

REFERENCES

(1) Snyder, N. M., Effect of air rate, temperature and packing density in a cross-flow cooling tower, American Institute of Chemical Engineering, Heat Transfer Symposium, vol 52, no. 18, 1956, p 61-67.
(2) Zivi, S. M., Brand, B. B., An analysis of the cross-flow cooling tower, Refrigerating Engineering, Aug. 1956.

PRESIDENT'S PAGE
J. DONALD KROEKER
Guest Sponsor

Toward Questions That Cannot B

The Long Range Planning Committee has come into the Society's limelight recently as a result of its study and report on research of the Society, detailed in the January issue of the JOURNAL (page 55) which has led to a change in the method in which the Society will pursue its research in the future.

As a result of this report and the Society's action pursuant thereto, a need has become apparent for information on what this Committee is, its functions, and its place in the Society structure. This has resulted in a request by President Tull to discuss this Committee and its work.

In past years, both ASRE and ASHAE have had special committees assigned to the study of specific subjects of a forward-looking or long-range nature.

A continuing Long Range Planning Committee by that name was established by ASHVE at the Semiannual Meeting of that Society in June, 1953. In any society, situations arise which become apparent to the Board of Directors as being less than ideal or even being improper. The problems involved are often minor ones but sometimes are inextricably interwoven within an entire field of operation of the society.

For example, there might be and have been questions of methods of raising funds for research. In itself the subject is fraught with many facets. However, it cannot be studied alone, since it might affect the entire research program, the income from Society publications, the amount of dues, or other important areas of Society operation. Another example is the withdrawal of a student branch at an educational institution. This raises a question as to the advisability of having student branches and requires study of an entire process and segment of the Society.

NOT ENOUGH FACTS

Often the Directors may not be equipped with enough facts to lead to logical solutions of such problems. Frequently, other phases of Society activities are involved. Further, in meetings of the Board of Directors, it is often difficult to find and maintain all applicable factors in perspective.

The present Long Range Planning Committee was established by the By-laws of ASHRAE to overcome these natural impediments to logical solutions of problems, to gather and organize data on all facets of sometimes complicated problems, and to suggest solutions to these problems so that the Board of Directors could take consistent effective action. The By-laws reads as follows:

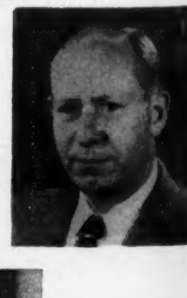
"The Long Range Planning Committee shall consistently make the necessary studies to prepare for and recommend to the Board of Directors long range planning on the aims and activities of the Society which in the opinion of the Committee would affect the future welfare and growth of the Society."

According to this section of the By-laws, the Committee may itself select its fields of operation. However, as specific problems, deemed to involve a larger phase of the Society or inter-related phases of the Society, become apparent to the President or to the Board of Directors, subjects have been and are referred to the Long Range Planning Committee for study and reporting.

The Committee has been assigned a number of studies and engaged in a num-

ASHRAE'S LONG RANGE PLA

J. DONALD KROEKER
CHAIRMAN



FRANK H. FAUST

P. B. GORDON

Be Studied Alone

ber of its own initiative. These have resulted in reports on research, administration, and public relations, which caused modifications of some features of the Society and in establishment of an active public relations program. A study on membership resulted in changes in membership grades, which now follow the pattern recommended by ECPD.

Prior to the merger, ASRE had its own monthly periodical. That of ASHAE was included as the JOURNAL SECTION in Heating, Piping & Air Conditioning. A report on publications of ASHAE, developed with the assistance of a publications consultant, explored the feasibility of publication of a separate Journal. Considered also at the time was the possibility of publishing an ASHAE Journal jointly with ASRE. About this time, a joint ASRE-ASHAE Committee on Cooperation was established, which explored community of interest and eventually led to renewing discussions of the feasibility of consolidation of the two societies, urged strongly as early as about 1933 or 1934.

A comprehensive study on finances of the Society was also made in 1955 and reported.

Probably the most widely known study and report of the Committee was that on regional operation of chapters, developed in 1955. This resulted in the present organization, in which the chapters have a voice in the Society through their chapters' agenda and through election of members to the Nominating Committee. About the same time, ASRE was restudying its organization of sections on a regional basis. It is interesting that, to a large extent, the regional operation of ASRE became parallel to and almost exactly that recommended by the Committee and established in ASHAE.

Another study and report of 1955 was that on student branches, which extended into surveys of faculty advisors to student branches in operation. Also, a survey of other institutions was made to determine the desirability of establishing further student branches. This resulted in a clear directive for a course which has proved logical.

Special branches and overseas branches were studied and reported in 1957 and policies developed which are believed still in effect.

In the late 1950s, it became apparent that efforts would be required to obtain a greater percentage of engineering graduates for one of the most rapidly expanding industries in history; namely, air conditioning. Approaches and methods were developed in a study titled "Engineering Manpower."

The organization of ASHAE was essentially that established in the 1920s and had been brought up to date only piecemeal, resulting in imbalance of emphasis and capacity. To determine an organization which would best accomplish the aims and objectives of ASHAE, a comprehensive study was undertaken by the Committee in 1957. The report was not adopted, since consolidation of ASRE and ASHAE was being considered. However, the report was used extensively in studying and evaluating the organization required and established under ASHRAE.

The composition of the Committee through the years is also interesting. The Committee consists of six members and has been in operation consistently for eight years. A total of 17 have served on the Committee.

PERSONS OF EXPERIENCE

It is obvious that a committee of this kind, making studies for future trends and accomplishments of the Society, must be composed of persons who have had wide and deep

(*Continued on page* 102)

PLANNING COMMITTEE

DAN D. WILE
VICE CHAIRMAN

ELMER R. QUEER

S. J. WILLIAMS, JR.

Pike's Peak, Colorado's best-known mountain, is seen here in snow-capped 14,110-ft serenity from a vantage point in the Garden of the Gods, near Colorado Springs.

Denver will be next June 26-28

An across-the-city-to-the-mountains view, showing Denver's Civic Center. In the foreground, the City and County Building, as seen from the west steps of the Colorado Capitol Building.

Once again, ASHRAE heads to a scenic center for its summer meeting.

Region IX and the Rocky Mountain Chapter thus become host for the three-day 68th Annual Meeting.

Program details and social events are still in the making; meanwhile here are three pictures made available by the local committee which should whet your appetite to be in Denver June 26-28 and to anticipate such points of interest during your stay.

Switzerland-in-America is what they call the 100-mile stretch of Highway 550 as it ranges north from Durango to Ridgway over 10,-910-ft Molas Divide and 11,018-ft Red Mountain Pass.

Bypass method simplifies prediction of

Spray Coil Performance

with heated spray water

The air side performance of a spray coil unit, at winter design, can be determined in a distinct and certain manner using a cooling tower technique. Two methods described herein provide a ready solution to spray coil performance with heated spray water and a better understanding of the basic phenomena underlying the unit's operation.

By utilizing this new approach to the subject, the product design engineer may more easily prepare realistic ratings while the application engineer will have available a clear cut means of making an accurate and total solution. But a single determination of water side performance and the use of two newly derived formulas are required.

The second procedure relates to the use of a bypass factor which may be calculated from the same water side performance. This simple bypass factor concept may then be used to determine the air side performance in still another manner.

When such a unit is selected for summer design conditions, it does not follow that satisfactory space conditions are maintained at winter design. A cooling load, at winter design conditions, demands a dry bulb temperature of air entering the space lower than space design, yet of a moisture content which produces design humidity. Since neither the air side nor the water side performance can exist independently, it follows that assumed values for the entering or leaving spray water temperatures, to and from the unit, cannot provide a satisfactory basis for calculating the condition of the air leaving the spray section. Saturation efficiency is meaningless when applied to simultaneous heating and humidification.

William T. Lyons is a member of the Application Engineering Dept. of the Carrier Air Conditioning Company, Division of Carrier Corporation.

WILLIAM T. LYONS

Actually, a parallel exists between the operation of a spray coil unit and a cooling tower, so that both may be analyzed using the same approach. Such a parallel becomes evident as the following factors are considered:

(1) In a spray coil unit, with heated spray water, air simultaneously is heated and humidified and in the process the spray water is cooled.

(2) In a cooling tower water is cooled and in the process the air, used to cool the water, simultaneously is heated and humidified.

(3) Thus, the following relationship is set up:

	Spray coil unit with heated spray water	Cooling tower
Purpose of process	To heat and humidify air	To cool water
Incidental effect	Water is cooled	Air is heated and humidified

The above table illustrates how the purpose of the spray coil unit becomes the incidental effect of the cooling tower and the purpose of the cooling tower becomes the incidental effect of the spray coil unit.

(4) In each case, purpose and incidental effect are mere technicalities, so that in essence the phenomena involved are the same.

In both cases, the air and water side performances are utterly dependent upon each other such that neither, for a moment, can exist independently.

A spray coil unit, as shown in Fig. 1, sometimes is referred to as a surface type humidifier. Water drawn from the basin, below the extended surface coil, is picked up by the recirculating pump and discharged to a spray water heater.

From the heater the water is pumped through piping and nozzles, where it is sprayed into the air stream and wets the coil. In the process the water is cooled and the air is heated and humidified. The extended surface of the coil serves the same purpose as the fill in a cooling tower, which is to provide additional water film surface for

By using the bypass factor method it is unnecessary to determine the value of the spray water temperature beyond a single determination of water spray side performance.

The intimate contact of air and water effects an interdependence of air and water side performances which the bypass factor method recognizes.

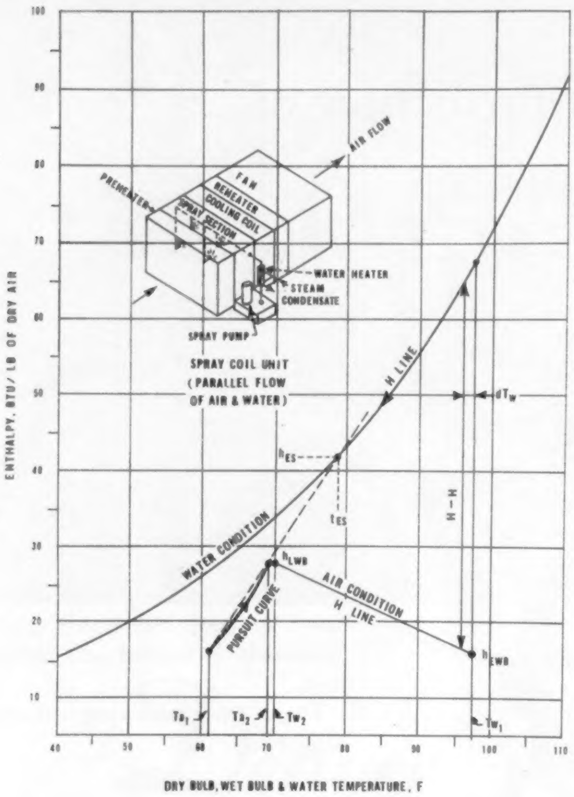


Fig. 1 Performance spray coil unit with parallel flow of air and water

the desired evaporation of water.

Referring again to Fig. 1, the saturation line H' represents the water condition as it passes through the spray section of the unit (including coil) as well as the enthalpy of the air at the surface of the water film. This line is the same as is found on a psychrometric chart. The H line represents the air condition (enthalpy) as it passes through the spray section, in the same direction as the water (parallel flow).

The temperature difference between the spray water entering and leaving the spray section (T_{w1}-T_{w2}) is analogous to the "range" of a cooling tower and is equal to the temperature rise given it in the spray water heater. The pursuit curve represents the dry bulb temperature of the air as it passes through the spray section of the unit.

Actually, the state of the air describes a curve which is a pursuit curve of the state of the water film surface. Such a curve may be constructed on an enthalpy – temperature diagram with the aid of the "water condition" and "air condition" lines, in the same manner as it would be constructed on a psychrometric chart but with greater simplicity. The dry bulb temperature of the air leaving the spray section "Ta_2" may also be calculated. In either case the value of the spray water temperature entering the unit must be determined.

As is the case with a cooling tower, the performance pattern of a spray coil unit is fixed so that only a physical alteration of the unit or a change in the water or air quantity through the unit will change it. A single determination of such performance and the use of Formula #1 are all that is required to determine the spray water temperature entering the unit, for a given entering air wet bulb temperature. Formula 2 then is used to calculate the dry bulb temperature of the air leaving the spray section. The proper use of these formulas constitutes the first method of solving for spray coil performance with heated spray water.

The necessity for the use of logarithms in Formulas 1 and 2 (which follow) has been eliminated by the inclusion of the curves in Figs. 2 and 3, respectively, where the power of "e" is the abscissa and the right side of the equation is the ordinate.

Formula #1

$$e^{N_s X} = \left\{ 1 - [.0362\, R/(Y + X) \right\} / \left\{ 1 - [.0362\, R/(Y - X)] \right\}$$

Formula #2

$$e^{(Ns/27.6)[(\text{air rise}/K) - 50.23 + 3101.14\ \text{gpm/cfm} + Ta_1]} = [(.0181)\,(TW_1)^2 - 1.82\, TW_1 + H] / [.0181\,(TW_1 - R)^2 - 1.82\, TW_1 + R\,(1.82 - 112.36\ \text{gpm/cfm}) + H]$$

(The right-hand side of the equation in Formula #2 is equal to the enthalpy difference between the H' line and the H line at TW_1, divided by the corresponding difference at TW_2—see Fig. 1.)

Where:

$X = \left\{ ([112.36\ \text{gpm/cfm}] - 1.82)^2 - .0724(H - [112.36\ \text{gpm}\ TW_1/\text{cfm}]) \right\}^{1/2}$

$Y = .0362\ TW_1 + (112.36\ \text{gpm/cfm}) - 1.82$

R = temperature difference (F) between spray water entering and leaving spray section or spray water heater capacity Btu/hr/500 gpm

Ns = Spray section performance factor (constant) = No. of transfer units or $\int_{TW_2}^{TW_1} d\,TW/(H' - H)$

gpm = gal/min of spray water circulated

Table I—H Values for Various Wet Bulb Temperatures of Air Entering Unit

Wet-bulb Temperature F	H*	Wet-bulb Temperature F	H*	Wet-bulb Temperature F	H*
40	57.89	51	52.26	61	45.97
41	57.42	52	51.68	62	45.27
42	56.95	53	51.10	63	44.55
43	56.46	54	50.50	64	43.81
44	55.97	55	49.90	65	43.06
45	55.47	56	49.28	66	42.29
46	54.96	57	48.64	67	41.50
47	54.44	58	48.00	68	40.70
48	53.91	59	47.34	69	39.87
49	53.37	60	46.66	70	39.03
50	52.82				

* H = 73.12-h EWB

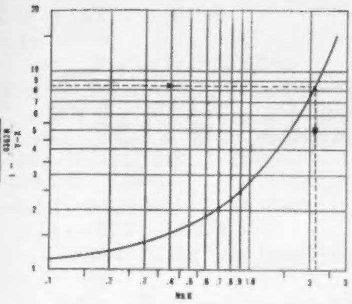


Fig. 2

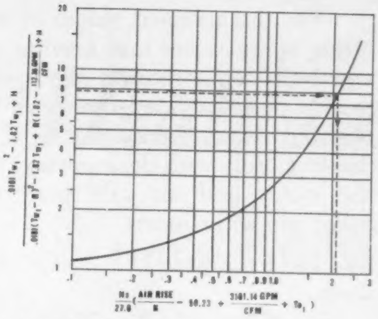


Fig. 3

Fig. 2-5 Graphic solutions to related equations used in accompanying text

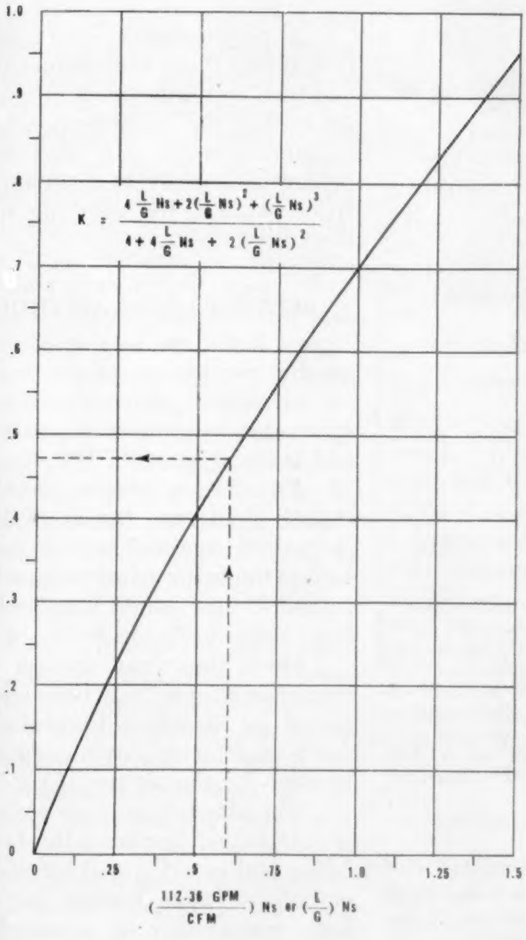


Fig. 5

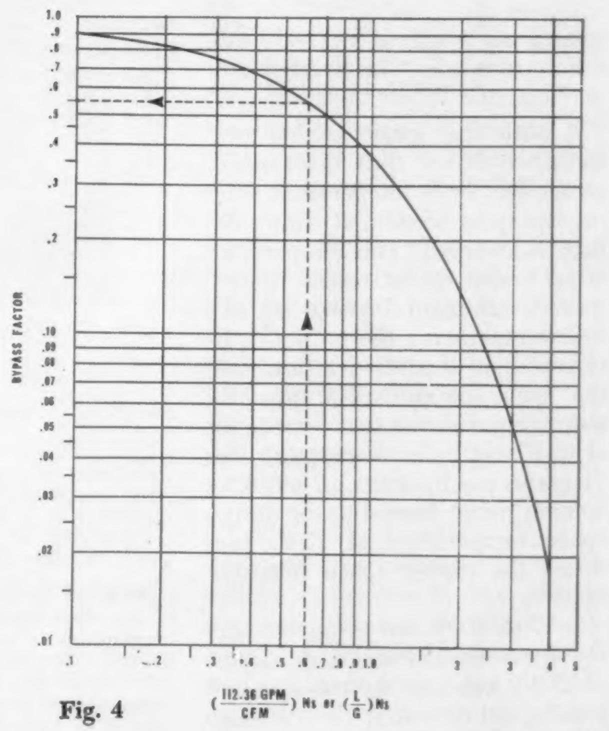


Fig. 4

cfm = cu ft of air/min through spray section and coil

air rise = temperature difference (F) between air entering and leaving spray section (including coil)

K = constant, from curve in Fig. 4, dependent upon value of (112.36 gpm/cfm) Ns

Ta_1 = air temperature (F) entering spray section

TW_1 = water temperature (F) entering spray section

H = value from Table I, dependent upon wet bulb temperature of air entering spray section

Proper use of the formulas should be qualified: (1) The formulas are applicable for spray section entering water temperatures at or below 110 F and leaving water temperatures at or above 70 F (2) The water and air quantities must remain constant for any set of calculations.

Example 1 – A spray coil unit, with spray water heater, has been selected for space conditions of 75 F dry bulb temperature and 55% relative humidity, year round. The required air quantity, at summer design, has been determined to be 5000 cfm and will be used year round. Because of a cooling load down to an outside winter design temperature of 10 F, the supply air temperature to the space must not exceed 70 F dry bulb to maintain a space temperature of 75 F. 100% outside air is supplied to the space.

Recirculating spray water pump capacity is 19 gpm. Spray water heater capacity is 260,000 Btu/hr. Manufacturers' original tests on unit prototype indicate that with 5000 cfm through the unit and with the spray water heater at 100% capacity the spray water is cooled from 110 to 82.6 F with an entering air wet bulb temperature of 64.5 F.

What will be the dry bulb temperature of the air leaving the spray section at winter design and what will be the required capacity of the preheater?

The spray section performance factor should first be determined by means of Formula #1.

Solution #1—Part 1

Given:

R = 27.4 = (110 F — 82.6F)

TW_1 = 110 F

gpm = 19

cfm = 5000

H = 43.44 from Table I for 64.5 F wet bulb temperature of air entering spray section

'Required: Ns

Procedure:

Solve for "X" and "Y" and then use Formula #1 to solve for Ns

$$X = \left\{ [(112.36(19)/5000] - 1.82)^2 - 0.0724\ (43.44) - [112.36(19)\ 110/5000]) \right\}^{1/2} \quad (1)$$

$$= (1.94 + .31)^{1/2} = 1.5 \quad (2)$$

$$Y = 0.0362\ (110) + [112.36\ (19)/5000] - 1.82 \quad (3)$$

$$= 3.98 - 1.39 = 2.59 \quad (4)$$

$$e^{NsX} = \left\{1 - [0.0362\ (27.4)/(2.59 + 1.5)]\right\} / \left\{1 - [0.0362(27.4)/(2.59 - 1.5)]\right\} \quad (5)$$

$$= [1 - (0.992/4.09)]/[1 - (0.992/1.09)] \quad (6)$$

$$= 0.76/0.09 = 8.51 \quad (7)$$

From Fig. 2, $NsX = 2.13$ (8)

From step 2, $X = 1.5$ therefore $Ns = 2.13/1.5 = 1.42$ (9)

With the spray section performance factor thus determined, an air wet bulb temperature leaving the spray section, at winter design, is assumed. This temperature is such that, at or below the required minimum leaving air dry bulb temperature, the proper moisture content is assured to maintain the space humidity. A psychrometric chart shows that leaving air at 62 F wet bulb temperature and 71 grains per lb of dry air moisture content, when heated to the design space temperature of 75 F, produces the design space humidity of 55%.

With 5000 cfm of air through the spray section and a difference of 27.4 F between the entering and leaving spray water, the enthalpy difference (Btu/lb dry air) between the entering and leaving air is 11.7 or (112.36 gpm/cfm) R. Since the leaving air wet bulb temperature is assumed at 62 F (27.85 Btu/lb dry air) the entering air wet bulb temperature is 42 F (16.15 Btu/lb dry air).

An entering spray water temperature of 97.4 F is assumed. With a difference of 27.4 F between the entering and leaving spray water temperatures, the leaving spray water temperature (TW_2) is 70 F.

This allows for an approach of 28 F to the entering air wet bulb temperature of 42 F as compared to an approach of 18.1 F to the entering air wet bulb temperature of 64.5 F in the manufacturers' original tests, Part 1. For the same spray section performance factor the approach increases with a decrease in the entering air wet bulb temperature.

For the assumed values of entering spray water and leaving air wet bulb temperatures, the spray section performance factor must be checked using Formula #1. If it checks closely with the known value the water and air side performances are as assumed.

Solution #1—Part 2

Given:

$R = 27.4 = (97.4\ F - 70\ F)$

$TW_1 = 97.4\ F$

$gpm = 19$

$cfm = 5000$

$H = 56.97$ from Table 1 for 42 F wet bulb temperature of air entering spray section

Required: Ns

Procedure:

Solve for "X" and "Y" and then use formula #1 to solve for Ns

$$X = \left\{ ([112.36(19)/5000] - 1.82)^2 - 0.0724\ (56.97 - [112.36(19)\ 97.4/5000]) \right\}^{1/2} \quad (1)$$

$$= (1.94 - 1.11)^{1/2} = 0.91 \quad (2)$$

$$Y = 0.0362\ (97.4) + [112.36\ (19)/5000] - 1.82 \quad (3)$$

$$= 3.53 - 1.39 = 2.13 \quad (4)$$

$$e^{NsX} = \left\{ 1 - [0.0362(27.4)/(2.13 + 0.91)\right\} / \left\{1 - [0.0362(27.4)/(2.13 - 0.91)]\right\} \quad (5)$$

$$= [1 - (0.992/3.04)]/[1 - (0.992/1.22)] \quad (6)$$

$$0.67/0.19 = 3.54 \quad (7)$$

From Fig. 2, $NsX = 1.27$ (8)

From step 2, $X = 0.91$ therefore, $Ns = 1.27/0.91 = 1.40$ (9)

With the spray section performance factor checked, the air rise (F) may be resolved by means of Formula #2. A psychrometric chart shows that outside air at 10 F (100% humidity), when heated to a wet bulb temperature of 42 F, leaves the preheater and enters the spray section at a dry bulb temperature of 61.1 F (Ta_1).

Solution #1—Part 3

Given:

$TW_1 = 97.4\ F$

$R = 27.4 = (97.4\ F - 70\ F)$

$H = 56.97$ from Table 1 for 42 F wet bulb temperature of air entering spray section

$gpm = 19$

$cfm = 5000$

$Ns = 1.42$

$Ta_1 = 61.1\ F$

$K = 0.47$ from curve in Fig. 4 for a value of 0.61 for (112.36 gpm/cfm) Ns

Required: Ta_2 and required capacity of preheater

Procedure:

Use Formula #2 to solve for air rise (F)

$$e^{(Ns/27.6)[(\text{air rise}/K) - 50.23 + (3101.14\ gpm/cfm) + Ta_1]}$$

$$= \left\{0.0181\ (97.4)^2 - 1.82\ (97.4) + 56.97\right\} / \left\{0.0181\ (97.4 - 27.4)^2 - 1.82\ (97.4) + 27.4\ [1.82 - (112.36(19)/5000)] + 56.97\right\} \quad (1)$$

$$= (171.71 - 177.27 + 56.97)/(88.69 - 177.27 + 38.17 + 56.97) \quad (2)$$

$$= 51.41/6.57 = 7.82 \quad (3)$$

From Fig. 3, the power of "e" = 2.08 (4)

Therefore, $(1.42/27.6)\ [(\text{air rise}/0.47) - 50.23 + 3101.14(19)/5000 + 61.1] = 2.08$ (5)

$$\text{air rise}/0.47 = [2.08\ (27.6)/1.42] + 50.23 - 11.79 - 61.1 \quad (6)$$

$$\text{air rise}/0.47 = 40.23 + 50.23 - 11.79 - 61.1 \quad (7)$$

$$\text{air rise} = 17.64\ (0.47) = 8.2 \quad (8)$$

$$Ta_2 = Ta_1 + \text{air rise} = 61.1 + 8.2 = 69.3\ F \quad (9)$$

The required capacity of the preheater is 5000 (1.08) (61.1 — 10) or *275,940 Btu/hr*

A psychrometric chart shows that at 69.3 F dry bulb temperature and 62 F wet bulb temperature the air leaving the spray section has a moisture content such that, when heated sufficiently to maintain the space at 75 F, the resulting space humidity is 55%.

BYPASS FACTOR METHOD

Bypass factor can be considered to be that percentage of the leaving air which has passed through the spray section completely unaltered and is equal to $e^{-(L/G)Ns}$.

Fig. 5 is a graph on which (112.36 gpm/cfm) Ns or $(L/G)N_s$ is plotted against bypass factor. L/G is the water to air ratio, where L and G are water flow and air flow, respectively, in lb/hr.

Once the spray section performance factor has been determined by means of Formula #1, the bypass factor may also be used to solve for the air rise (F).

The effective surface enthalpy is determined first from the bypass factor and entering and leaving air enthalpies. (The leaving air wet-bulb temperature is assumed, as was done in Solution 1, Part 2.) After converting the effective surface enthalpy to an effective surface temperature by means of the saturation line in Fig. 1 (or the ASHRAE GUIDE), the air rise (F) is resolved by means of the bypass factor, the effective surface temperature and entering air temperature. The effective surface enthalpy or temperature should be considered as the uniform surface enthalpy or temperature of the water film which produces the same leaving air conditions as the non-uniform enthalpy or temperature which occurs in operation.

(*Continued on page* 69)

Flow and Heat Transfer

characteristics of finned tube exchangers

Heat transfer and friction loss characteristics of a surface are determined by the exact nature of the flow process over the surface. There are two basically different kinds of flow possible near the surface of a solid, separated and unseparated flow. In unseparated flow fluid moves parallel to the surface and high velocities are found quite close to the surface. In separated flow the region of well ordered flow is removed from the surface, and intervening space is filled with a fluid of varying velocity moving about in a complicated way.

Unseparated flow occurs, for example, on a flat plate placed in a stream at zero angle of attack. Both separated and unseparated flow are found on blunt bodies, such as a cylinder or plate normal to the flow.

Various measurements of local heat transfer on surfaces having regions of separation have indicated that heat transfer rates are not vastly different for separated and unseparated portions of the surface.[1] Heat transfer rates for a long cylinder and for a plate at zero angle of attack are essentially equal for ordinary fluid velocities, if heat transfer areas are the same per ft of span, if $\pi D = 2L$.

The principal effect of separation is to cause flow losses. Separation produces a region of low pressure on the back side of the surface and results in a high form drag. Total drag, which is an indication of flow losses, is the sum of frictional and form drag. For a cylinder and typical flow conditions in air, the form drag is over 90% of the total. For a zero angle of attack plate there is no form drag;

B. GEBHART

all flow losses are due to frictional forces. Therefore, cylinder flow losses are of the order of ten times as great as those of the plate with the same area per ft of span.

Comparisons of relative heat transfer and flow losses demonstrate that separation is an undesirable effect, if flow losses are important. The point of view taken in this work is that high heat transfer rates and low flow losses are both important.

EFFECTIVENESS RATIO

Effects of fluid separation indicate a need for a quantity which combines heat transfer and flow loss characteristics of a surface into a single parameter. A number of methods have been introduced in the past to permit such comparisons. See, for example, the discussion in Ref. (2). However, since these parameters do not combine the heat transfer and friction loss characteristics in a simple, direct way, a quantity called "effectiveness ratio" was defined.

The effectiveness ratio is heat transfer effectiveness divided by flow loss effectiveness. Heat transfer effectiveness, ε_E, is the amount of heat transferred divided by the max amount that an ideal exchanger would transfer, given the fluid temperatures.

$$\varepsilon_H = \frac{U_o\, A\, \Delta t_h}{m\, c_p\, \Delta t_m}$$

The flow loss effectiveness, ε_E, is defined as the flow energy dissipation rate divided by the kinetic energy of the fluid streams. In this study of air flow over finned tubes, ε_E is based upon air side losses alone.

$$\varepsilon_E = \frac{\Delta p_f}{\dfrac{\rho_a\, V^2}{2g_o}}$$

The effectivenesses are dimensionless, as is their ratio N, the effectiveness ratio.

$$N = \frac{\varepsilon_H}{\varepsilon_E}$$

The effectiveness ratio depends upon the heat exchanger geometry and, in general, is a function of the fluid flow Reynolds number and the fluid Prandtl number.

It appears to be a good indicator of the desirability of a geometric arrangement. Streamlined surfaces generally have small separated regions and, therefore, high values of N. Blunt or bluff bodies have large separation regions and small values of N, as shown in Table I. These values are based

TABLE I. EFFECTIVENESS RATIO FOR THE FLOW OF AIR OVER ISOTHERMAL SURFACES OF VARIOUS GEOMETRIES

Surface	N	Flow Condition
Flat Plate at zero angle of incidence	0.63	Laminar Boundary Layer
	0.63	Turbulent " "
Turbulent Flow inside a tube	0.70	All Reynolds numbers
Flow normal to a cylinder	0.141	$N_{Re} = \frac{U_o D}{\nu} = 100$
	0.072	$= 1{,}000$
	0.018	$= 10{,}000$
	0.009	$= 100{,}000$

B. Gebhart is Associate Professor of Mechanical Engineering, Cornell University. This paper, here somewhat condensed, was presented at the ASHRAE Semiannual Meeting in Chicago, February 13-16, 1961. The complete version is planned for inclusion in the 1961 ASHRAE TRANSACTIONS.

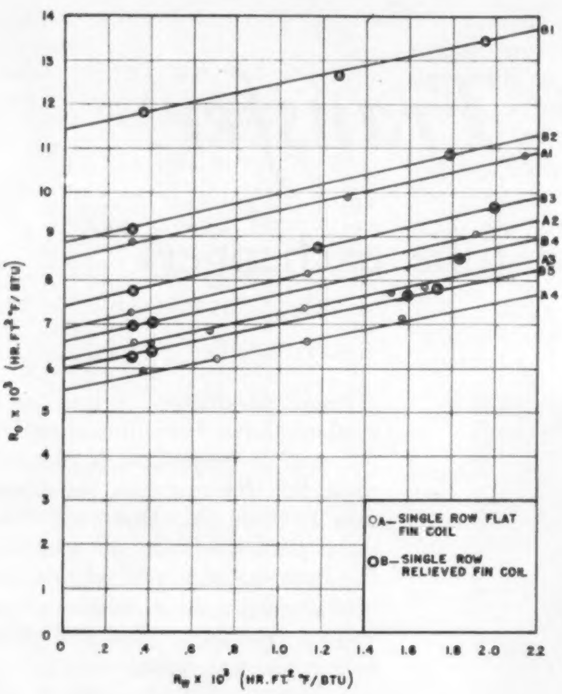


Fig. 1 Overall resistance vs water side resistance series A & B (for single coil, flat and relieved fins)

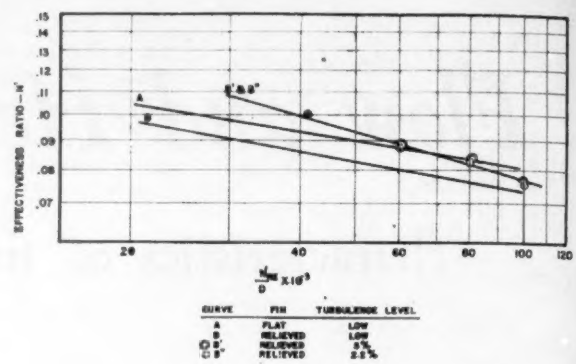


Fig. 2 Effectiveness ratio dependence upon Reynolds number

upon actual heat transfer and pressure drop information obtained from standard references. The flat plate at zero angle of attack is a streamlined form and causes no separation at all. Turbulent flow inside a tube has no separation. Values for a circular cylinder are much lower, due to separation.

Similar calculations, based on the performance of several types of deformed and relieved fin coils, show that the effectiveness ratio is also a sensitive indication of the flow pattern over this type of exchange surface. A flat fin has a relatively low N, undoubtedly because a portion of the fin surface contributes only friction losses. Slots in the fin do not cause important separation and a high value of N is found. Rippling of the fin would be expected to cause separation and results in a low value of N.

VISUALIZATION OF FLOW PATTERNS OVER FINNED TUBE EXCHANGERS

Since the nature of the flow pattern over a heat transfer surface is of crucial importance in determining relative heat transfer and flow loss characteristics, a study of the flow pattern was first carried out. Models were studied in a visualization tunnel by using smoke filaments.

Models of finned tube coils were constructed. These studies were designed to show whether flow separation on the fin causes deformed fins of the rippled type to be poorer than fins relieved by slotting. Models of the good (slotted) and the poor (rippled) deformed fins were built and studied for evidence of separation and boundary layer relief.

Visual observations with smoke filaments indicated that the flat and slotted fins produce no separation and that the rippled fin has extensive separation behind every sharp corner. The various smoke observations may be summarized as follows:

(1) Back half of the tubes were always separated flow.
(2) Fin surface in the wake of the tube is an area of high turbulence.
(3) Flow over all other flat fin surfaces is smooth and unseparated.
(4) Separation occurs over all sharp corners.

CONSIDERATIONS IN DESIGNING RELIEVED FINS

The preceding sections indicate observations and arguments whereby the point of view adopted was that a good deformed fin design is one which maximizes the effectiveness ratio. Although a large value of N indicates specifically high heat transfer with low pressure loss, it may also imply certain other things as well, for example, that unseparated flow produces little noise. Separation and consequent vortex and turbulence production result in greater noise.

The builder of the exchange surface is vitally concerned with many aspects of extended surface which have nothing directly to do with effectiveness ratio (for example, low material cost, simple construction, and low maintenance). However, since decisions on the relative importance of such factors are necessary to obtain a truly optimum practical design in each specific circumstance, and because the relative weights of the various factors change from one case to another, a simpler point of view was adopted in this study. A high effectiveness ratio was sought using fin designs which require no more material than the plain flat fin and which are as simple to assemble. Better fin shapes seemed possible and practical within these limitations.

RELIEVED FIN DESIGNS AND TESTS

Many particular fin types were considered. The fins tested were designed to provide compact, low flow loss, high heat transfer rate surface. Separation was to be avoided.

Coils having six different relieved fin designs were built, tested, and compared with the flat fin coil in the air velocity range from 200 to 1000 fpm. Tests were carried out in a draw-through tunnel. The coil specifications and fin types are illustrated in Fig. 6. The tube and

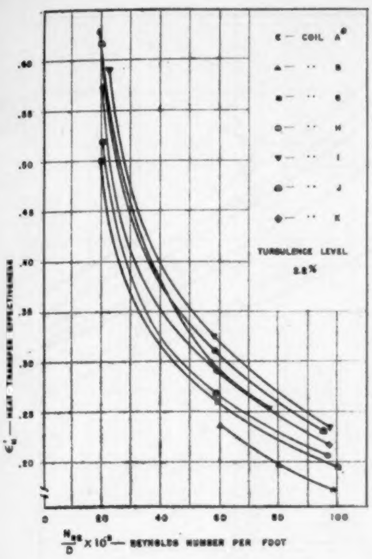


Fig. 3 Heat transfer effectiveness vs flow Reynolds number. *Undetermined low turbulence level for coil A

fin materials and methods of manufacture and assembly were the same for all coils described.

Coil A is the flat fin against which comparisons were made. Coil B is the first relieved fin design and is based upon punching out reliefs in a flat fin. The fin areas farthest from the tubes were removed. Slots and holes were provided to interrupt the fin boundary layer. The fin requires only 75% as much material and has only 64% as much area as the plain flat fin.

Coils A and B were tested in a low turbulence air stream at air velocities of 200, 400, 600 and 800 fpm in the following six arrangements:

A—Single flat fin coil
B—Single relieved fin coil
C—Two flat fin coils in line, spacing 1 in. between coil centers
D—Two relieved fin coils in line, spacing 1 in. between coil centers
E—Two flat fin coils staggered, spacing 1 in. between coil centers
F—Two relieved fin coils staggered, spacing 1 in. between coil centers

Test Series A and B were a comparison of the two fins in a low turbulence stream. In Series C and D the disturbances produced by the first row tubes mainly encounters the second row tubes and in Series E and F mainly encounters the second row fins.

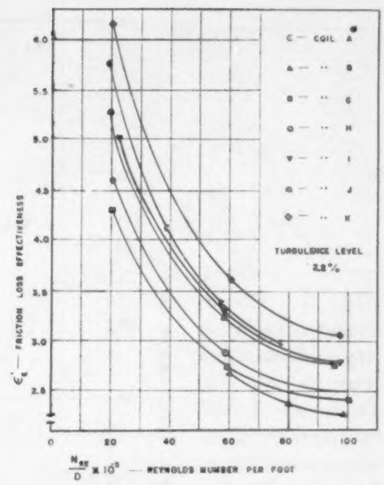


Fig. 4 Flow loss effectiveness vs flow Reynolds number *Undetermined low turbulence level for coil A

The nature of the boundary layer relief and the results of the tests of the multi-row arrangements of Coils A and B suggested that the relieved fin would benefit from turbulence in the air stream. Since considerable turbulence is usually present in actual heat exchangers, the single relieved fin coil was tested at air velocities of 400, 600, 800 and 1000 fpm for two turbulence intensity levels.

The draw-through tunnel was modified by placing several layers of fine mesh screen over the inlet and by locating a turbulence producer in the upstream section. The turbulence producer was a crossed rod grid made of ¼-in. wood dowelling on 1¼-in. centers.

The turbulence level produced by the grid at the location of the coil under test varies with the distance between the grid and coil. The decay curve for a grid having a diam-spacing ratio of 0.2 as established by Dryden[3] and later investigators was the method used. There is some uncertainty in the use of these results in a different set-up, but the agreement of various investigators of turbulence decay supports the procedure. Compare with the unpublished results of Davis discussed by Sato and Sage.[4]

Turbulence intensity, i, may be defined as the root-mean-sq average of the turbulent disturbances which is expressed as a percentage of the bulk velocity of the stream.

The flat fin was not re-run; it is expected that the turbulence levels employed would have little effect on its performance.* Two series of tests were carried out for a single relieved fin coil. The first series, labelled B′, had a turbulence intensity of 5.0%; the second series, labelled B″, had an intensity of 2.2%.

Results of the tests outlined in the preceding paragraphs guided the design of five additional relieved fins. Each fin was designed to test a specific effect.

The five designs, designated G, H, I, J, and K are illustrated in Fig. 6. Fin G is merely fin B with the trailing edge relief and forward holes omitted. Fin H is fin G with the back holes omitted. Fin H was designed to test the postulate that, due to boundary layer build-up, the area between the tubes on the trailing edge of the fin would have relatively low fluid friction and be an area of relatively high temperature. The value of the forward holes was doubted, but the back holes were left in fin G to check the effect of boundary layer interruption in the cylinder wake.

The remaining three fin designs (I, J, and K) were different surface relief patterns applied to a fin with a slightly less pronounced leading edge relief. It was thought that the leading edge relief for B (and for G and H as well) was perhaps excessive and that some effective heat transfer area had been removed. Compare leading edge dimensions for coil B and I in Fig. 6.

Coil I has the most boundary layer relief. A new boundary layer begins on the raised portion and on the fin surface behind the relief. In Coil J there is a single relief. The fin design for Coil K was an attempt to reduce flow losses by reducing flow separation behind the cylinder. It was hoped that the raised tabs would direct flow in behind the cylinder and that reduction in flow losses would more than offset the reduction in heat transfer which would result from eliminating the heat transfer area.

Each of the five new fin types

* This may be inferred from the work reported in Ref. 4 and 5. For example, in Ref. 4 intensities of the order of 5.0% produced a change in the heat transfer coefficient of only 3% on a sphere.

(G, H, I, J, and K) were tested in a single coil arrangement in the draw-through tunnel at turbulence intensity levels of approximately 2.2% and 5.0%. For 2.2% turbulence, each coil was tested at 200, 600 and 1000 fpm air velocity and for 5.0% turbulence, at 600 fpm air velocity.

RESULTS

Coils and arrangements indicated in the preceding section were tested under the various air velocity and turbulence conditions stated. For each test condition the coil performance was extrapolated to zero water side heat transfer resistance (i.e., "infinite" water velocity). This permitted a comparison of behavior which did not include the effect of the water side convection coefficient.

The quantity R_o was extrapolated to obtain R'_o. R_o was based upon the same area for all coils, i.e., the inside tube surface area. Calculations are outlined in the Appendix.

For each test several runs were taken at different water velocities. The value of R_o was calculated from the measured heat transfer rate and plotted against water side resistance R_w, calculated from Equation 3 in the Appendix. The resulting curve should be a straight line of slope 1.0 since:

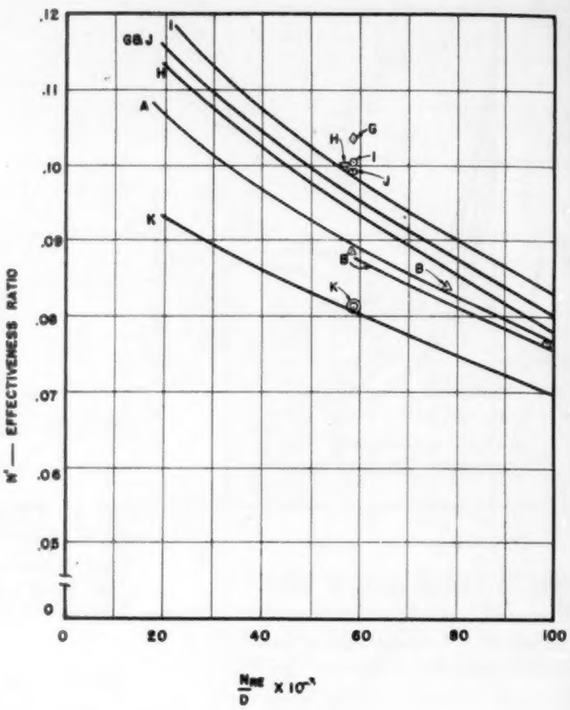
$$R_o = R_a + R_m + R_w \text{ and } \frac{dR_o}{dR_w} = 1.0$$

The spread of the points from a straight line was extremely small, due, apparently, to excellent heat balances. The slopes of all the best fit curves are within a few deg of the average of their slopes, which was 1.05. See, for example, Fig. 1.

All curves were subsequently drawn at this slope. For some of the later tests, measurements were taken for only one water flow rate, the highest which permitted sufficiently accurate measurement of water side temperature change.

The flow loss characteristic for each air velocity and arrangement was calculated by methods outlined in the Appendix. Values for various water velocities at a given air velocity were extrapolated to zero water side resistance. The curves had an approximate zero slope, as expected.

Fig. 5 Effectiveness ratio vs Reynolds number (curves are for Tests at 2.20% turbulence intensity, except for A. Points are for 5.0% turbulence intensity)



Data for the low turbulence test of single coils A and B are plotted and extrapolated in Fig. 1. Number designations (A1, A2, etc.) indicate the approximate air velocity level. The number is multiplied by 200 to obtain the velocity in fpm. The extrapolated values, R'_o and ε'_E, and the calculated values of ε'_H and N' will be listed in TRANSACTIONS. The value of ε'_H is calculated from the last quantity of Equation 4 of the Appendix.

Results for the six arrangements A, B, C, D, E and F show many interesting features. For flat fin coils, the two row staggered arrangement has the lowest resistance, followed by the single row. Relieved fins behave in the same way. This difference may be caused by the disturbances of the first row tubes passing over the second row fins.

The relieved fin resistance is higher than that of the flat fin by 30, 30 and 20% for the single coil, in-line, and staggered arrangements. Had the resistance been based upon fin area, the relieved fin coils would have been better by 15, 15 and 25%. The writer feels, however, that a comparison should be made on the basis of tube surface.

The foregoing discussion of the values of R'_o applies also to the heat transfer effectiveness ε'_H, since it is determined directly from R'_o.

Flow loss of the relieved fin is much lower than that of the flat fin in all arrangements. Therefore, relieving has produced an undesirable effect upon resistance and a desirable effect upon flow losses. The effectiveness ratio, which shows the net effect, is plotted in Fig. 2 for the single coil arrangements A and B. The decrease in flow loss does not offset the increase in resistance. The flat fin is better in all arrangements, the difference being, 10, 5 and 5% for the single coil, in-line, and staggered arrangements.

The superior performance of the flat fin in these tests is apparently due to two reasons. First, calculations indicate that a .008-in. fin thickness is close to optimum for the air velocity range studied. Therefore, fin edges are still quite effective heat transfer areas, and slots on the edges of the relieved fin remove much effective area. Second, the tests were carried out in a low turbulence tunnel. Although both fins would benefit from turbulence, it appears certain that the relieved fin would benefit more. Turbulence would disturb the boundary layer as it passes over the slots and holes.

Tests were carried out at turbulence intensity levels of 2.2 and 5.0% for a single, relieved fin coil. These results were extrapolated to zero water side resistance. The re-

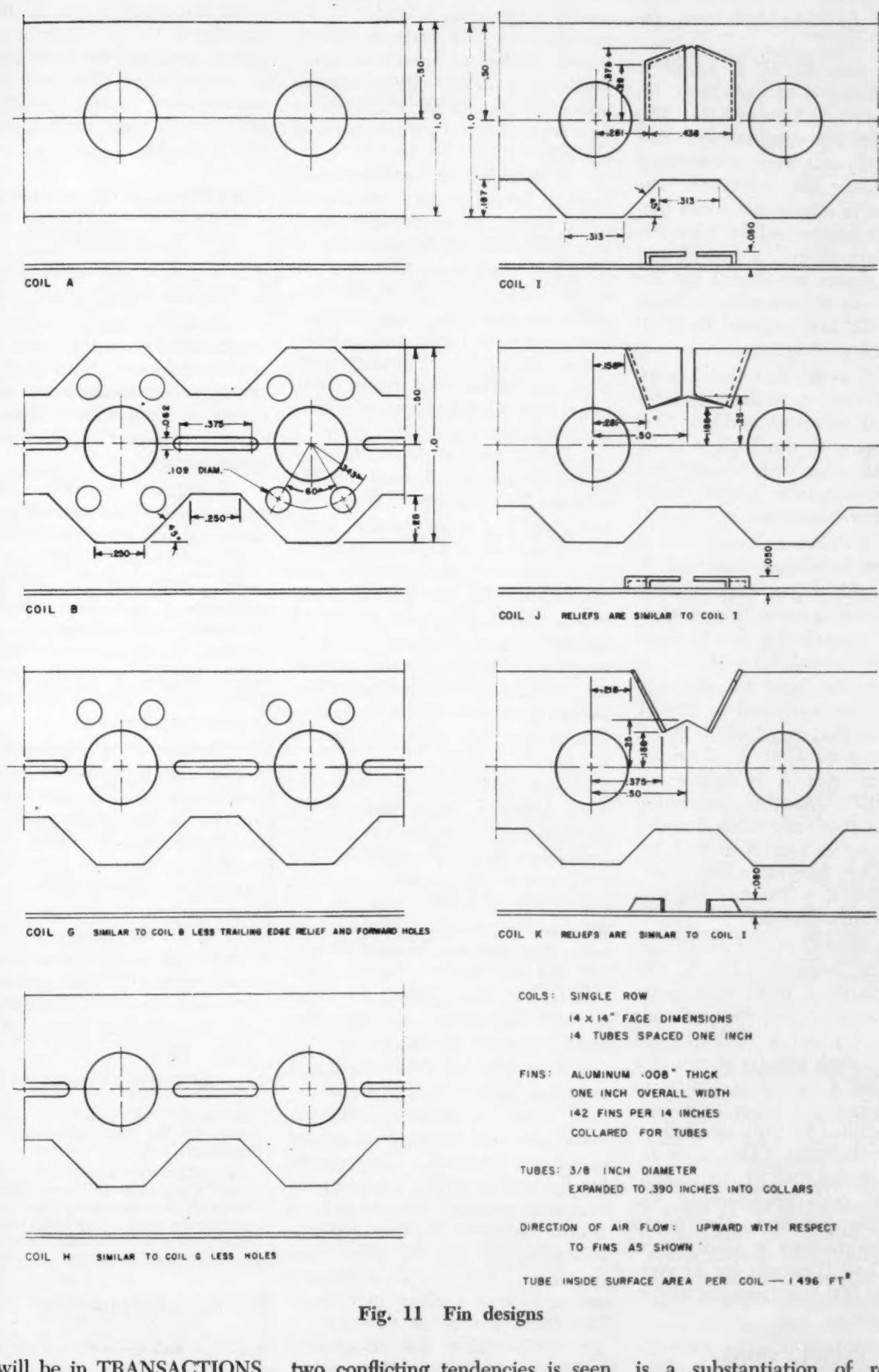


Fig. 11 Fin designs

sults will be in TRANSACTIONS.

Turbulence reduces flow loss and enhances heat transfer for the relieved fin. Resistance to heat transfer is still higher than that for a flat fin, but the pressure loss is much less. The net effect of these two conflicting tendencies is seen from a comparison of N′ in Fig. 2. Turbulence has substantially improved the performance of the relieved fin coil and has made it somewhat better than that of the flat fin coil at low velocities. This is a substantiation of reasoning which predicted such a result.

However, improvement is not sufficient to consider this relieved fin better than the flat fin. Given thickness of the fin material, this design is not superior, since con-

siderable fin area has been removed.

The coils G, H, I, J and K were also tested at turbulence intensity levels of 2.2 and 5.0%. The heat transfer resistance and flow loss effectiveness were extrapolated to zero water side resistance. The heat transfer effectiveness and friction loss effectiveness for 2.2% turbulence are plotted in Figs. 3 and 4. Also shown are curves for the flat fin (Coil A) at low turbulence and for the first relieved fin (Coil B) at 2.2% turbulence.

Fig. 3 shows that Coil I is superior to Coil A in heat transfer rate for all velocities and that Coil J is superior in the high velocity range. All other coils transfer less heat, per unit tube length, under similar flow conditions.

Fig. 4 shows that only Coil K has higher flow losses than Coil A. That is, all designs other than K have lower flow losses than A. Failure of K is evidently due to separated flow behind the tabs.

Effects of heat transfer and flow loss are combined in Fig. 5, and we see that only Coils B and K are inferior to Coil A. The improvement over A is largest for Coil I, 15%. Another interesting feature is that only Coils I and J are superior to Coil A in both respects. They give more heat transfer and less flow loss per coil, i.e., per ft of tube length.

The effect of higher turbulence levels is also shown in Fig. 5. The 5.0% turbulence tests were made at approximately 600 fpm for coils B, G, H, I, J and K. Extrapolated results for each coil are shown as a point. Coil K, which is similar to a flat fin, in that it offers no boundary-layer relief, is little affected by higher turbulence. (This is an indirect corroboration of the contention that turbulence would not benefit the flat fin.) The group which are superior to Coil A (i.e., G, H, I and J) benefit by an average 4% at 600 fpm from the higher turbulence.

CONCLUSIONS

Surfaces having separated and unseparated flow have comparable heat transfer rates, but separation results in much higher flow losses. The effectiveness ratio, defined in this study, evaluates these effects and is a sensitive indicator of the performance of a surface. Visualization studies of models of cylinders and of finned tube exchangers indicated conditions which cause flow separation on various types of surfaces.

A relieved fin, which would provide boundary-layer relief and avoid separation on the fin surface was designed. Coils, made up of such fins, were compared over the air velocity range 200 to 800 fpm with flat fin coils. The comparisons, made for single and two-row inline and staggered arrangements in a low turbulence stream, indicated that fin relief was excessive in this design.

Initial tests with the first relieved fin design indicated that the relieved fin benefitted from disturbances in the air stream. Since turbulence of a high intensity is often present in actual equipment, the relieved fin was tested at turbulence levels of approximately 2.2 and 5.0%. Turbulence substantially increased heat transfer and reduced the flow losses for this coil. The effectiveness ratio under these conditions somewhat exceeded that of the flat fin.

Five additional relieved fins were designed. They were tested as single coils in streams having turbulence levels of approximately 2.2% and 5.0% intensity. At 2.2% turbulence intensity, four of the five fins had a higher effectiveness ratio than the flat fin, and two of the fins had both a higher heat transfer rate and a lower flow loss. At 5.0% turbulence, the improvement was even greater.

This series of observations and tests has shown that rational design, based upon flow considerations, can lead to much improved exchanger surface. The specific flat fin against which the comparisons were made is believed to have almost optimum thickness, width, and tube spacing for these flow conditions. Yet several of the second group of designs give more heat transfer with less flow loss.

Improvement was 15 to 20% for the best designs. However, it would be unrealistic to assume that this amount of improvement is a limit for optimum relieved fin designs in actual use in exchangers. The writer feels that considerably better designs are possible. Another important factor is that the fins tested in this study were designed as much for low flow loss as for high heat transfer. Greater improvement seems possible, if only one of these two characteristics is sought.

ESTIMATES OF ACCURACY AND RELIABILITY OF THE RESULTS

The accuracy and reliability of the test results depend upon the magnitude of the errors in the measurements taken during the tests. There are three sources of error: fluctuations (or unsteady state effects), errors in temperature determinations, and errors in flow rate determinations.

Long and carefully controlled warm up periods insured approximately steady state conditions during the time in which data were taken. Since all but one of the variables of first order effect were instantaneous readings, the period of a run was quite short. Observed drifts during the period of a run indicated an error of at most 1.0% in heat transfer rates.

Estimates of inaccuracies in the air side temperature difference and mass rate of flow determination indicate a maximum error in the computed heat flow rate of 2.0%. Similar estimates for the water side suggest a maximum error of 3.3%. Therefore, the max expected difference in the two heat flow rates for a given run is 5.3%. Max difference found was 4.0% and occurred in only one run. The average difference, for all runs, was 2.0%.

ACKNOWLEDGMENTS

This study was carried out under the sponsorship of the Salvatore Giordano Foundation.

The author appreciates the assistance of Messrs. Klaus Conrad, Felix J. Pierce, and Robert L. Putman, who conducted the heat transfer tests and reduced the results. Thanks are also due to Mrs. Leora Decker who typed the manuscript.

REFERENCES

1. Eckert, E. R. G. and Drake, R. M. Jr., "Heat and Mass Transfer," 2nd Ed., McGraw-Hill (1959).
2. McAdams, W. H., "Heat Transmission," 3rd Ed., McGraw-Hill (1954).
3. Dryden, H. L., et al., NACA Report 581 (1937).
4. Sato, K., and Sage, B. H., "Thermal Transfer in Turbulent Gas Streams," Trans. ASME, Vol. 80, 1958, p 1380.
5. Kestin, J., and Maeder, P. F., "Influence of Turbulence on Transfer of Heat from Cylinders," NACA TN 4018 (1957).

NOTATION

U_o — Overall surface coefficient from water to air
R_o — $1/U_o$
R'_o — $R_m + R_a = 1/U'_o$ — The value of R_o extrapolated to zero water side resistance
U'_o — $1/R'_o$
R_m — Heat transfer resistance of tube wall and fin material
R_a — Heat transfer resistance between fin surface and air stream
R_w — Heat transfer resistance of water side
ε_E — Flow loss effectiveness
ε'_E — ε_E at zero water side resistance
ε_H — Heat transfer effectiveness
ε'_H — ε_H at zero water side resistance
N — Effectiveness ratio
A_i — Inside tube surface area of a coil $= 1.496\ ft^2$
c_p — Specific heat at constant pressure
m — Mass rate of flow in lb/sec
t_a — Air temperatures
t_w — Water temperatures
Δt_h — Heat transfer temperature difference
Δp_s — Static pressure change of air through heat exchanger
Δp_f — Frictional effects expressed as a pressure decrease
H_L — Frictional head loss (in.H_2O)
V — Air velocity ahead of heat exchanger
ρ — Density
μ_a — Air viscosity
N_{Re} — Reynolds number

Subscripts

1 — Ahead of heat exchanger
2 — After heat exchanger
a — Air
w — Water
h — Denotes Δt upon which U_o is based
m — The temperature change for an ideal exchanger
am — Average air temperature
wm — Average water temperature

APPENDIX

Calculation Procedures

Outline for single and double coil arrangements

$$R_o = \frac{1}{U_o} = \frac{A_i \ \Delta t_h}{q} \quad (1)$$

$$\Delta t_h = \frac{(t_{w1} - t_{am}) - (t_{w2} - t_{am})}{1n \dfrac{t_{w1} - t_{am}}{t_{w2} - t_{am}}} \quad (2)$$

$$R_w = \frac{1}{h_w} = \frac{4.14 \times 10^{-4}\, m_w^{-0.8}}{(1 + .011\, t_{wm})} \quad (3)$$

from $N_{Nu} = .023 (N_{Re})^{.8} N_{Pr}^{1/3}$

$$\varepsilon'_H = \frac{U'_o A_i \Delta t'_h}{m_a c_p (t_{w1} - t_{a1})} \approx \frac{U'_o A_i}{m_a c_p} = \frac{A_i}{c_p m_a R'_o} \quad (4)$$

where the prime denotes infinite water velocity

$$\Delta p_f = \Delta p_s - \Delta p_m \quad (5)$$

$$\Delta p_s = \Delta H_m \rho_{G1}$$

$$\Delta p_m = \frac{m \Delta V}{A_i g_o} \approx \left(\frac{\rho_1 V_1^2}{2g_o} \right) \left(\frac{2 \Delta t_a}{T_{a1}} \right)$$

$$H_1 ('H_2O) = \frac{\Delta p_f}{\rho_w}$$

$$\varepsilon_E = \frac{\Delta p_f}{\dfrac{\rho_1 V_1^2}{2g_o}}$$

$$H_L (Iso.) = H_L \frac{\rho_1}{\rho_m} = H_L \frac{T_{am}}{T_{a1}} \quad (6)$$

$$N' = \frac{\epsilon'_H}{\epsilon'_E} \quad (7)$$

Air flow conditions upstream of the heat exchanger may be generalized by referring to a Reynolds number per ft of heat exchanger dimension.

$$\frac{N_{Re}}{L} = \frac{\rho_a V}{\mu_a}$$

SPRAY COILS

(*Continued from page* 62)

The equation for the effective surface enthalpy is as follows:

hES = hEWB + [(hLWB — hEWB)/(1 — B.F.)]

The equation for air rise (F) is as follows:

air rise (F) = (1 — BF) (tES — Ta_1)

Where:

hES = effective surface enthalpy (Btu/lb dry air)
hEWB = enthalpy of air entering spray section (Btu/lb dry air)
hLWB = enthalpy of air leaving spray section (Btu/lb dry air)
B.F. = bypass factor
tES = effective surface temperature (F)
Ta_1 = temperature of air entering spray section (F)

Example 2 – From Solution 1, Part 1, the spray section performance factor (Ns) was calculated to be 1.42. Determine bypass factor and solve for the dry-bulb temperature of the air leaving the spray section, at winter design, using the two preceding equations.

Solution #2

Given:

hEWB = 16.15 Btu/lb dry air (ordinate) from the saturation curve in Fig. 1 for a value of 42 F wet bulb temperature (abscissa).
hLWB = 27.85 Btu/lb dry air (ordinate) from the saturation curve in Fig. 1 for a value of 62 F wet bulb temperature (abscissa).
gpm = 19
Ns = 1.42
Ta_1 = 61.1 F
cfm = 5000
R = 27.4

Required: Ta_2

Procedure:

Determine bypass factor from Fig. 5 and use the two preceding equations to find air rise (F)

(112.36 gpm/cfm) Ns = [112.3(19)/5000]1.42 = 0.61 (1)

From Fig. 5 for a value of 0.61 for (112.36 gpm/cfm) Ns the bypass factor = 0.55 (2)

hES = 16.15 + [(27.85 — 16.15)/(1 — 0.55)] (3)
= 16.15 + 11.7/0.45 = 42.15 Btu/lb dry air (4)

From the saturation curve in Fig. 1 for a value of 42.15 Btu/lb dry air (ordinate)

tES = 78.6 (F) (abscissa) (5)

air rise (F) = (1 — 0.55) (78.6 — 61.1) (6)
= 0.45 (17.5) = 7.9 (7)

Ta_2 = 61.1 + 7.9 = *69.0 F* (8)

Note that in Fig. 1, a line drawn through Ta_1 and Ta_2 on the pursuit curve and extended to the saturation line bisects the latter at a point which, when projected downward to the abscissa, will equal the effective surface temperature (F) tES.

Note also that by using the bypass factor method it is unnecessary to determine the value of the spray water temperature, entering the unit, beyond a single determination of water side performance.

In the case of a spray coil unit, as in a cooling tower, the intimate contact of air and water effects an interlocking of air and water side performances. Accordingly, logic would indicate that an approach used to predict water side performance might also be utilized to predict air side performance and vice versa. Such reasoning is justified amply, especially in Solution #2 where the bypass factor $e^{-(L/G)Ns}$ is used to predict air side performance and in Solution #1, Part 2, where Ns is used to predict water side performance.

. . . simply because they have not been

There has been a great deal of information published and reported on the design of ventilating and air conditioning systems but there has been conspicuously little reported on field testing and its relation to the basic design of the final installation.

Many installations of perfectly good design, provided with first-class equipment, often do not produce satisfactory results simply because they have not been field tested and adjusted.

One of the final responsibilities of the consulting engineer is to supervise the testing of the major equipment and the adjustment of the air flow in the duct system.

Prior to this stage, the problems and requirements for the system will have been studied thoroughly, criteria established and drawings and specifications prepared to indicate clearly the intent for the installation. Complete design performance data for all equipment, such as fans, pumps, dehumidifiers, filters, heating coils, refrigeration machines, cooling tower, and other related component parts of the system will have been included in the specifications.

The contractor installed the system in accordance with the drawings and specifications, but the equipment should be field checked for performance to produce the desired end result for which it was designed. During the course of installation, the contractor submitted for approval shop drawings of all main apparatus rooms, ductwork and major equipment. For the major equipment, the contractor also submitted for approval certified performance data. The importance of shop drawings for the duct work cannot be over-estimated as they often show deviations from the contract drawings which might influence the adjustment, testing and balancing of the final installation.

Field testing of the air conditioning or ventilating installation is, to some extent, a dual responsibility of the contractor and the consulting engineer, but the end result of the system is the responsibility of the engineer. The contractor is obligated to prove the specified performance of the installed equipment and to balance the air throughout the duct system.

For ventilating or air conditioning systems, provisions should be included in the design for checks and tests upon the performance of the major components. As all parts of the system function to affect each other, all must be field checked for conformity with the specified design requirements. For example, flow meters in the chilled and condenser water piping circuits will measure the quantity of water being circulated when testing the performance of the refrigeration machine, cooling tower, cooling coils and main chilled and condenser water pumps.

Where directed by the engineer, reliable pitot tube stations should be installed throughout the ductwork to balance the air for the design volumes to the branches and terminals. Balancing of the air throughout the duct system must be completed before the fan performance can be checked. Thermometers and pressure gauges should be installed to measure the temperatures and pressures in and out of the condenser and the evaporator of the refrigeration machine and to measure the pumping heads imposed on the main pumps.

Consider a typical air conditioning installation consisting of, in the direction of air flow, minimum and maximum outside air dampers, return air dampers, mixing plenum, main filter bank, spray washer coil type dehumidifier, dehumidifier air dampers, bypass filters, reheater and damper, main supply fan and discharge ductwork. Consider also that the system is provided with a return-relief fan which either returns air from the conditioned space to the main apparatus or discharges the air through a relief air damper, depending on the dictates of the automatic temperature control.

SPECIFIC PROCEDURES

General inspection of the installation should be made to make certain that all splitter or quadrant dampers are not in a position to block air flow completely. The fan drive should be checked for rotation, speed and belt slippage and adjustments made, if required. Power readings should be taken on the motors to make certain they

Here is an authoritative discussion of those specific procedures which a consulting engineer should follow in field checking and testing some of the major components of any ventilating or air conditioning system.

All other equipment, forming the complete installation should be checked with the same degree of accuracy that the system, as a whole, will produce satisfactory results.

James Bricker, Vice President, Charles S. Leopold, Inc., Engineers, presented this paper as "Field Check and Testing of Ventilation and Air Conditioning Systems" at the Ventilation Symposium, ASHRAE Semiannual Meeting, Chicago, Ill., February 13-16, 1961.

Field Tested and Adjusted

JAMES BRICKER
Member ASHRAE

are not overloaded. The conditioning apparatus on the suction side of the supply fan should be checked for proper installation of the main filter bank to protect the cooling coil, for freedom of operation of all the automatic dampers, and for water level in the dehumidifier to assure passage of air through the cooling coil.

As mentioned previously, before the air cycle can be tested for performance of the fans and adjusted for the design air flow through the conditioning apparatus, the supply fan discharge ductwork must be balanced for the specified air volumes to the various branches and terminals. The discharge duct work should be analyzed to determine which branch or terminal is the most critical. Usually, but not always, the critical branch duct or terminal is the most remote of the duct system. This selected branch or terminal should be used as the base for the adjustment of the entire fan discharge duct system.

For brevity in this discussion, consider that the supply fan discharge duct, as shown in Fig. 1, is extended and sub-divided into four branches with the first branch being the nearest to the fan and the fourth branch being the most remote.

For the balancing of the air being delivered, a pitot tube and draft gauge should be used for quantitative measurements. The use of the pitot tube and draft gauge for this purpose is generally considered acceptable by most standards. In each branch, pitot tube stations should be provided downstream of the adjustment damper so that it is not always necessary to take a complete traverse of velocity pressures for every movement of the damper while balancing each branch.

If for each velocity pressure traverse the static pressure is also recorded, the square law can be used for any further adjustments until the desired static pressure required for the designed volume of air has been reached.

At the station for branch #4, in Fig. 1, a pitot tube traverse of velocity pressures should be taken and the static pressure also recorded. With this information, the volume of air being supplied to this branch may be calculated. Using this amount of air pro rated for the design quantities as required, the outlets or terminals on this branch may be balanced relatively to each other. For this adjustment a velometer, anemometer or similar instrument can be used. Having balanced the terminals relative to each other, retake a complete traverse at the pitot station and recalculate the volume of air. If under design, adjust its balancing damper for the static pressure required. In the event the required pressure cannot be reached, even with a fully open damper, then partially close the balancing damper in branch #3 until the static pressure for the design volume of air in branch #4 is indicated on the draft gauge.

Having established the design volume of air and the static pressure in branch #4, a draft gauge should be connected to measure this pressure continually as a base requirement while the remaining branches are adjusted and balanced. Constant reference to this base static pressure should be made as the system is balanced progressively towards the fan.

Before any movement of the balancing damper in branch #3 is made, a preliminary traverse of velocity pressure and static pressure should be made and the volume of air calculated. Pro rating the air, balance the terminals on this branch relative to each other as previously described for branch #4 and retake a complete traverse including a static pressure reading.

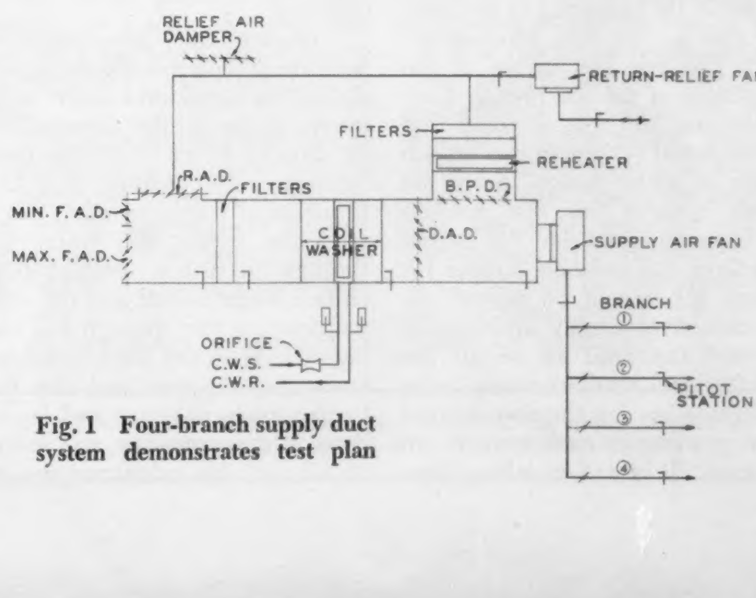


Fig. 1 Four-branch supply duct system demonstrates test plan

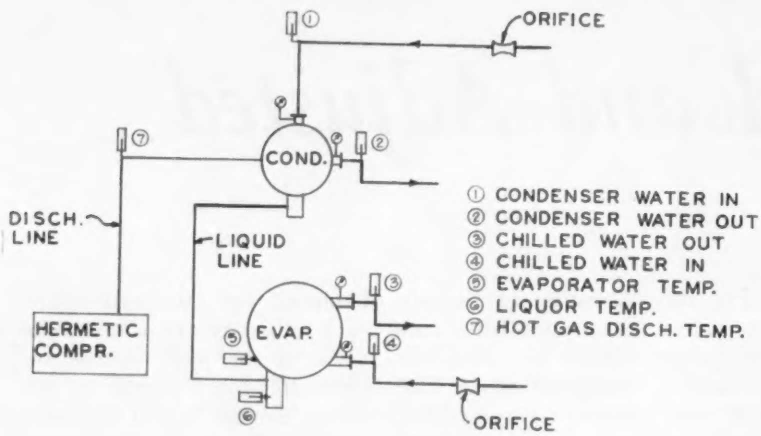


Fig. 2 Hermetic compressor with cooling tower

If this traverse indicates insufficient air flow, the balance damper in branch #2 should be adjusted to force more air towards branches #3 and 4.

This procedure may increase the base static pressure to branch #4 which will necessitate readjustment of its balance damper until the base pressure is reproduced. Repeat a traverse at branch #3 and if it is at the design requirements proceed with the balancing of the next branch, #2, towards the fan. As you proceed towards the fan, the base static pressure can also be moved towards the fan to a point more convenient for constant reference as the remaining system is balanced. For example, the new base static pressure could be relocated just after the connection at the main duct for branch #2 and the static pressure indicated at this location would be the pressure required for the volume of air to both branches #3 and 4.

The same procedure should be used in the balancing of branches #2 and 1. As you proceed towards the fan and arrive at the adjustment of the first branch nearest the fan, you may discover that the measured volume in this branch is over or under design. Since the system beyond this first take-off has been balanced for the design quantities, the balance damper for branch #1 cannot be moved. In this case, it is usually desirable to pro rate the total air of all the branches and readjust them, using the square law for the new desired static pressure in each branch. In this event, it is well to take a complete traverse of velocity pressures for the last branch to make sure this readjustment is satisfactory.

As a check against the total air as measured for each individual take-off, a pitot tube traverse should be taken in the main fan discharge duct. The static pressure should also be noted and recorded.

The next step in the balancing of the duct system before the fan performance can be tested is the adjustment of the air flow through the conditioning equipment on the suction side of the supply fan.

With the bypass damper in a tight closed position, the dehumidified air damper can be adjusted for the design flow through coil washer by measuring the fan discharge static pressure and calculating the volume using the square law. When properly adjusted, measure the static pressure drop across the coil washer. Start opening the bypass damper slowly until the static pressure in the fan discharge duct indicates the value required for the design supply quantity.

During this adjustment the pressure drop across the coil washer should be noted and slight movements made to the dehumidified air damper to maintain the previously measured pressure drop. Confirmation of this adjustment can be made by noting the water flow through the orifice installed in the chilled water circuit and the water temperature rise through the cooling coil. With this information and measuring the wet and dry bulb temperatures entering and leaving the coil spray washer, the volume of air can be calculated on the basis of a heat balance. This will also check the cooling coil performance.

To check the amount of outside air and return air, repeat the static pressure drop across the coil washer from which the volume can be determined and measure wet and dry bulb temperatures of the outside air, the return air, and the mixture air. With these data, calculate the amount of fresh air and return air. In taking wet and dry bulb temperatures in the mixing plenum, the average of the several readings over the face area should be used, as sometimes there is stratification of air through the equipment.

The testing and balancing of the air flow through the return air system should be adjusted in the same manner outlined for the supply air system.

Our procedure for balancing the supply air through the fan discharge duct system is somewhat different from the usual method of beginning at the fan and working towards the end of the duct system. However, since the resistance to flow is calculated for the delivery of the air to the least favored terminal, it is apparent that the system should be balanced to produce the pressure at this terminal or branch for its design volume of air. Having completed the balancing of the fan discharge ductwork and the adjustment of the air flow through the conditioning equipment, the performance of the fan can now be checked.

For this purpose, repeat measurements of the static pressure in the fan discharge, at the fan suction plenum, and the static pressure drop across each major part of the conditioning equipment may be made. Recheck the power input to the motor and calculate the brake horsepower. These data should be compared with the design requirements and with the approved fan performance data submitted by the contractor.

Testing of the refrigeration cycle for the performance of the component parts should be conducted when the operating conditions are as near to the design conditions as possible. The purpose of this test is to determine any variation between the specified and the delivered performance.

COMPRESSOR-TOWER TESTS

Now consider the test procedure for an installation using a hermetic centrifugal compressor and cooling tower, as indicated on Fig. 2. The requirements for testing this type of refrigeration are similar for any type of refrigeration cycle as far as the basic information being sought is concerned.

The accuracy of the test is not dependent on the length of the run but, rather, on a stable operating condition which, more often than not, has to be imposed artificially in order to load the refrigeration machine. The load can be imposed on the evaporator by the use of more outdoor air through the cooling coils or by the use of the preheater coil, if available. Sometimes it is possible to make tests during a slow pull-down period after the plant has been shut down for a few hours.

Before proceeding with any test of the refrigeration cycle, the safety controls, overload controls, load limit control and safety trip control must be checked. An inspection should be made to assure the reliability of the test data to be recorded and should include the location of the various flow meters and manometers, the location of the various pressure gauges and the location of the various thermometers throughout the water circuits. In connection with thermometers, special attention should be given to the depth and contact of a thermometer stem in the wells in the piping to measure properly the water temperature and not the pipe or insulation temperature.

Usually, the chilled water temperature leaving the evaporator of the refrigeration machine is controlled automatically by a thermostat. For a steady state condition, the sensitivity of this instrument should be decreased to avoid the chance of the water temperature having too great an influence on the compressor operation during the test runs.

When testing the refrigeration cycle, it is not necessary to test all components at the same time. Usually it is physically impossible to do so. The compressor should be tested as a unit and the cooling tower as a unit. Log sheets should be prepared for recording test data to include the following: chilled water temperatures in and out of the evaporator and condenser water temperatures in and out of the condenser, compressor discharge and suction temperature, pressure as read on the gauges furnished with the compressor unit, liquor temperature, chilled and condenser water pump discharge and suction pressure, flow meter readings for the water being circulated through the evaporator and condenser, voltmeter and ammeter readings of the motor, condenser and evaporator pressures as measured by test manometer, motor overload relay settings, purge operation, oil pressure and position of inlet vane dampers.

For this, these instruments should be used: Laboratory tested, chemically etched glass thermometers to measure the various temperatures being recorded, a portable manometer for checking the evaporator and condenser pressures, a barometer, a clamp-on ammeter and accurate voltmeters and a watthour meter, if available, for checking readings to motor. Care should be taken when inserting the chemical thermometers in the wells. The wells should be packed with a high heat transfer compound so that there are no voids.

Having completed the preparation for the tests and having established a full load condition, a preliminary test of the refrigeration cycle should be conducted to establish what adjustments should be made for the subsequent test runs. This preliminary run is to establish, specifically, the condensing pressure for the design condenser water temperature leaving the condenser. Since the outside wet bulb will probably not be at the conditions specified, the water temperature leaving the condenser can be approached by reducing the quantity of water being circulated.

It is also important in testing the refrigeration machine that the chilled water return be at the design temperature to check the performance of the evaporator for the designed leaving chilled water temperature.

At least three or four sets of readings should be recorded as quickly and as accurately as possible. With these data, the evaporator tonnage and condenser tonnage can be calculated. From the power readings recorded, the horsepower should be determined and the heat of compression can be calculated for use in determining the condenser tonnage.

Results of the test should be analyzed and compared with the specified performance and with the performance data submitted by the compressor manufacturer, paying especial attention to the temperature difference between the water out of the condenser and the condensing temperature, and the chilled water temperature leaving the evaporator and the suction temperature. If the comparison is favorable, the refrigeration machine should be tested at reduced loads as, for example, 75 and 50% load.

The test of the cooling tower should also be conducted during a steady state operation. Before conducting this test, the fan motor amp and volt should be checked and the fan blade pitch adjusted, if required, to draw the contract horsepower. The water flow to the tower distribution pan should be checked for uniformity of distribution, the tower fill and eliminators should be cleaned to prevent channeling or obstruction of air flow. The specified water flow should be adjusted.

When testing the cooling tower, a number of wet bulb temperatures should be taken near the inlet louvers at the tower and at the tower fan discharge. A complete log of the refrigeration machine and the cooling tower should be recorded when operating the refrigeration machine under load.

For the major pumps, such as condenser and chilled water, measurements of the designed water flow through the orifice, the pressures on the suction side and discharge of each pump, the amp and volt, should be recorded and the horsepower input calculated. This result should be compared with the certified pump curves submitted for approval.

SUB-COOLING AND REHEATING ELIMINATED IN NEW DESIGN FOR INTERIOR OFFICE SPACE AIR CONDITIONING PAGE 41

Disposal of ASHRAE Laboratory Facilities and Equipment

E. P. PALMATIER

Chairman ASHRAE
Research and Technical
Committee

As announced in the January issue of the ASHRAE JOURNAL, the Society's Research Laboratory at 7218 Euclid Ave., Cleveland 3, Ohio, is being deactivated. Concurrently, steps are being taken to accelerate and expand sponsored work at universities and other laboratories.

One of the responsibilities of the Research and Technical Committee is the development of policies and procedures by means of which the facilities and equipment of the Laboratory may be liquidated. At first, this might seem like a simple task, but to do the job with proper consideration of the Society's research objectives and with complete fairness to potential recipients represents quite a challenge.

Actually, there are two parts to the problem. First, there is the question of relocating two major facilities which have been designed and erected for executing specific programs. Second, there is an inventory of general purpose instrumentation and equipment that might be of value to almost any laboratory.

MAJOR FACILITIES

One of these is the Controlled Environment Room, which has been used for the past few years to investigate the reaction of healthy subjects to various thermal and radiant environments. Negotiations are essentially completed, as this is being written, to relocate this facility at Kansas State University, where it will be used to carry on the program described in the January JOURNAL.

The second major facility is the Solar Calorimeter, which has been used for many years to study the solar heat gain through glass and other structural components, and the influence of shading devices on heat gain. Laboratories interested in acquiring this facility should address an inquiry to ASHRAE Research Laboratory, 7218 Euclid Ave., Cleveland 3, Ohio, to the attention of Assistant Director of Research Clark M. Humphreys. It will help to indicate the type of program contemplated for use of the facility, the staffing suggested procedures for working cooperatively with the Society (see April JOURNAL Research Page), the probable annual budget contemplated and the portion of this budget expected to be covered by grants from ASHRAE. The communication should also identify the individual with whom negotiations should be concluded and whether this person will be available at the ASHRAE 68th Annual Meeting in Denver on June 26-28. All inquiries will be evaluated by committee at that time and a recommendation will be submitted to the Board for action.

It is the intention to donate this facility, with its associated instrumentation, with the understanding that the recipient will handle rigging and trucking to its destination. Rigging to its present location on top of a one-story building was done with a small crane years ago at a cost of about $20. Today's cost might be $50. It can be shipped by flat-bed truck; weight is estimated at 1500 lb.

GENERAL PURPOSE INSTRUMENTS AND EQUIPMENT

An inventory has been prepared of this equipment, as shown on the facing page. Those interested in acquiring items should submit bids separately identified by item numbers to the attention of Mr. Humphreys at the ASHRAE Research Laboratory, 7218 Euclid Ave., Cleveland 3, Ohio. The deadline for consideration of all bids will be June 1, at which time the successful bidders will be notified. The equipment may then be picked up at the Laboratory or it will be packed and shipped at the bidder's expense price after June 15.

Any equipment that is not sold in this manner will be given away on a first-come-first-served basis. On July 1 the remainder will be sold in one lot or scrapped. Anyone wishing to inspect or evaluate the laboratory inventory may do so without prior notification.

Inventory of Laboratory Equipment

ELECTRIC MOTORS (60 cycle)

Item No.	Quantity	Manufacturers	hp	rpm	volt	phase
1	1	Allis Chalmers	2	1135	220	3
2	1	Allis Chalmers	3	1740	220/440	3
3	1	Emerson	½	1150	230	
4	1	Emerson	½	1725	110/220	1
5	1	Century	½	1735	220/440	3
6	1	Fairbanks	2	1800	220	3
7	4	G.E.	1/6	1140	115	1
8	1	G.E.	⅓	1725	220	3
9	1	G.E.	½	1100	220	3
10	2	G.E.	¾	1725	220/440	3
11	1	G.E.	3	1725	220/440	3
12	1	G.E.	5	1735	220/440	3
13	1	G.E.	7½	1800	220	3
14	1	G.E.	10	1740	220/440	3
15	1	G.E.	25	1760	220/440	3
16	2	Howell	⅓	1735	115/230	1
17	1	Wagner	1	1750	220	1
18	1	Emerald	1	1750	220/440	3

FANS AND BLOWERS

Item No.	Quantity	Manufacturer	Type	Size	Motor Data (60 cycle) hp	rpm	volt	phase	Remarks
19	1	American Blower	Sirocco	300					
20	1	American Blower		2½					C. I. Housing 12-in. Diam. Discharge
21	1	American Blower	Utility Set	30 H	¼	3000	115	1	
22	1	American Blower	Press. Blower	A	1/12	3450	115	1	
23	1	American Blower	Ventura	10 A			115	1	Kitchen Exh. Fan
24	2	American Blower	Ventura	100 C	¼	1750	115	1	V-Belt Driven
25	1	Buffalo Forge	HVA	151 A					
26	1	Buffalo Forge	Niagara Concidal	3½					
27	1	Buffalo Forge	CL	4					
28	1	Buffalo Forge	E	4½ E	1½	3475	220/440	3	
29	1	Buffalo Forge	E	5 E	7½	3520	220/440	3	
30	1	Bailey Blower	EX	55					
31	1	Clarage	CI	10					Slow Speed Wheel
32	2	Redmond	L	3876		1550	115	1	Cat. Rating 160 cfm
33	2	Emerson		89648			115	1	16-in. Man Cooling Fan
34	1	Emerson		89649			115	1	18-in. Man Cooling Fan

DIRECT CONNECTED MOTOR DRIVEN PUMPS (60 cycle)

Item No.	Quantity	Manufacturer	Type or Series	Size	Capacity gpm	Head ft	Motor Data hp	rpm	volt	phase
35	1	Bell & Gossett	Booster	1			1/6	1725	110	1
36	1	Deming	3903	1			1	3450	220/440	1
37	2	Deming	4003	1-A	15	37	½	1750	115/230	1
38	1	Eastern Industries	100	D-6			1/30	1550	115	1
39	1	Gorman-Rupp	210						115	1

SHELL AND TUBE HEAT EXCHANGERS

Item No.	Quantity	Manufacturer	Type	Size
40	1	Bell & Gossett	Cat. No. 150	Water Heater
41	1	Heat-X	CH538 36A1	Shell & Tube Chiller
42	1	Chiller for 7½ hp Compressor		

TOOLS

Item No.	Quantity	
43	1	Logan Lathe, Model 200, complete with accessories ½ hp, 220 volt, 60 cycle, 3 phase motor
44	1	Delta Bench Grinder, Model CD, Type SS ½ hp, 110 volt, 60 cycle, 1 phase motor
45	1	Delta 14-in. Bench Type Drill Press ⅓ hp, 115/230 volt, 60 cycle, 1 phase motor
46	1	Toledo Power Hack Saw ¾ hp, 220/440 volt, 60 cycle, 3 phase motor
47	1	Delta No. 34-307 10-in. Tilting Table Circular Saw 1 hp, 110/220 volt, 60 cycle, 1 phase motor
48	1	Oster No. 132 NV Power Drive ½ hp, 115 volt, 60 cycle, 1 phase motor
49	1	Stanley, Type 123A—½-in. Electric Drill—115 volt
50	1	Stanley, Type 324A—¼-in. Electric Drill—115 volt.
51	1	Stanley, Type W 65 Model A Electric Saw—115 volt
52	1	Mid-States Saw Gun
53	1	Niagara 36-in. Slip Roll Forming Machine No. 331
54	1	Grindstone with ¼ hp, 110 volt, single phase motor & belt drive
55	2	Machinist's Vises
56		Miscellaneous Hand Tools
57		Acetylene Welding Equipment
58	2	Handee Grinding Tool Sets
59	2	Work Benches

CONDENSING UNITS WINDOW AND ROOM AIR CONDITIONERS

Item No.	Quantity	Manufacturers	Type or Model	Motor Data (60 cycle) hp	rpm	volt	phase	
60	1	Carrier	7K3	3	1750	220/440	3	
61	1	General Electric	CM 63R	3	1725	220/440	3	
62	1	Westinghouse	RW 211	7½	860	220/440	3	
63	1	Carrier	51 Q2	½		115	1	Window Unit
64	1	Carrier	51 E2	½		115	1	Window Unit
65	1	Carrier	51 Q3	¾		230	1	Window Unit
66	1	Carrier	51 S3	1		230	1	Window Unit
67	1	Carrier	51 E4	1		230	1	Window Unit
68	1	Carrier	51 Q4	1		230	1	Window Unit
69	2	Chrysler	1675-8-39	¾		115	1	Window Unit
70	1	Philco	80R-11	¾		220	1	Window Unit
71	1	Philco	200FCW-2	2		230	1	Room Unit

MISCELLANEOUS EQUIPMENT

Item No.	Quantity	
72	2	Fultork Labmotors with Stirring Propeller, Fisher Scientific No. 14502
73	2	Immersion Heaters with Thermostats, 115 volt, 1000 watt, Fisher Scientific No. 14-463-10
74	1	Oven, Fisher Isotemp No. 130245, 230 volt, 550 watt
75	2	Vacuum Pumps, Central Scientific No. 90510 Pressovac 4, 110 volt, 60 cycle, single phase
76	1	Guarded Hot Plate, 8 x 8-in. Test Section, University of Minnesota Type
77	1	Vacuum Pump, Motoair, Size CD, 1/3 hp, 110 volt, 60 Cycle Motor
78	1	2 Wheel Truck, Howe No. 1-624
79	1	Gas Boiler, American Radiator Co. 1-GS-5
80	2	Speed Reducers, Ohio Gear Co., Cat. No. DOT-30A, Ratio 30-1
81	1	Set—8-in. Diam Sieves, 20, 60, 100, 140, 200, 230, 325 and 400 mesh, complete with receiver and cover
82	1	Speed Reducer, Boston Gear, No. LB900, Ratio 900-1
83	1	Air Compressor, Worthington Model 1657, Size 2 7/8 x 1 1/4, with 1/2 hp, 220 volt, 3 phase motor
84	1	Cooling Tower Marley Model 4340 with 1 1/2 hp, 1750/875 rpm, 220 volt, 3 ph., 60 cycle motor
85	1	24-in. Rotary Lawn Mower
86	1	Industrial Vacuum Cleaner, Premium Furnace Cleaner Model 175, 110 volt ac or dc, 1100 watt
87		4 1/2 lb Silver Solder (EntecRod 1800)
88	1	"15" Trojan Floor Scrubbing Machine
89	1	Concrete Mixer, Sears Roebuck
90	1	3 x 7-ft. Jamison Refrigeration Door
100	3	27 Drawer Steel Cabinets, 30W x 13D x 37 in. H, Olive Green
101	1	27 Drawer Steel Cabinet, 30W x 17D x 37 in. H, Olive Green
102	1	100 Drawer Cabinet, 36W x 9D x 34 in. H, Olive Green
103	12	Single Tier Steel Lockers, 12 x 18 x 72 in., Lyon
104	15	Single Tier Steel Lockers, 12 x 18 x 72 in., Supreme
105	1	36 ft Extension Ladder
106	1	24 ft Extension Ladder
107	1	14 ft Platform Ladder
108	1	12 ft Platform Ladder
109	1	10 ft Platform ladder
110	1	8 ft Step Ladder
111	1	6 ft Step Ladder
112	1	5 ft Step Ladder
113	3	Fire Extinguishers, 2 qt hand type
114	2	Fire Extinguishers, 15 lb CO_2
115	2	Aircoustal Units, Koppers Type STD-2-A
116	1	Bryant Air Dryer Model A-10, 1000-cfm capacity, Gas reaction, 95,000-Btu/hr input, 220-110 volt, single phase, 60 cycle

VARIABLE TRANSFORMERS

Item No.	Quantity	
117	2	Variacs, Type 200-CU, Cased, Output—0-135 volt, 7.5 amp
118	4	Variacs, Type 200-CU, Uncased, Output—0-135 volt, 7.5 amp
119	1	Variac, Type 50-A, Cased, Output—0-135 volt, 5 kva
120	1	Powerstat, Type 20, Output 0-135 volt, 0.4 kva
121	1	Powerstat, Type 116, Output 0-135 volt, 1 kva
122	1	Powerstat, Type 1126, Output 0-135 volt 2 kva

VOLTAGE STABILIZERS

Item No.	Quantity	
123	2	Voltage Stabilizers, Raytheon, Cat. No. VR6, Output 115 volt, 1000 watt
124	1	Voltage Stabilizer, Raytheon, Cat. No. VR7, Output 115 volt, 2000 watt
125	1	Sola, Cat. No. 30806, Output 115 volt, 120 volt-amp

Inventory of Laboratory Instruments

INSTRUMENTATION FOR AIR FLOW MEASUREMENTS

Item No.	Quantity	
126	1	Friez Thermo-Anemometer, No. A/50
127	1	Alnor Velometer, Type 3002, complete with 5 accessory jets
128	1	Alnor Velometer Type 3002, complete with 9 accessory jets
129	1	Hastings Air Meter, Model G 4
130	1	Hastings Air Meter, Model G 5
131	1	Alnor Thermo-Anemometer, Type 8500 with 24-in. probe, ranges 0-300 and 70-2000 fpm
132	1	Friez No. A/4Z Biram type revolving vane anemometer
133	1	Friez No. A/2Z Biram type revolving vane anemometer
134	2	Miscellaneous revolving vane type anemometers
135	1	Friez Cup Type Anemometer, Model No. 349 Cast aluminum nozzles of the following sizes: 1-1, 1-2, 1-3, 1-4, 2-6, 1-7, 2-9, and 1-10 in.
136	1	16 gage—4 1/2 in. diam Flanged Test Duct, complete with 1/2, 3/4, 1, 1 1/2, 2 1/2 and 3-in. brass orifices
137	1	16 gage—8 1/2 in. diam Flanged Test Duct, complete with 2 1/2, 3, 4, 5 and 6-in. brass orifices
138	3	Kata Thermometers (Red)
139	1	Kata Thermometer (Blue)
140	2	60 in. Pitot-Static Tubes, Revere No. R-361-PS
141	1	36 in. Pitot-Static Tube
142	1	30 in. Pitot-Static Tube, American Blower Type
143	1	24 in. Pitot-Static Tube, American Blower Type
144	4	Pitot-Static Probes, 0.058 x 12 in., Whitney Instrument Co.
145	2	Pitot-Static Probes, 0.095 x 12. in., Whitney Instrument Co.
146	2	Pitot-Static Probes, 0.095 x 16 in., Whitney Instrument Co.
147	2	Kiel Probes, 0.125 x 6 in.—Whitney Instrument Co.
148	4	Kiel Probes, 0.125 x 12 in.—Whitney Instrument Co.
149	2	Kiel Probes, 0.250 x 12 in.—Whitney Instrument Co.
150	3	Kiel Probes, 0.250 x 16 in.—Whitney Instrument Co.
151	1	Three Hole Yaw Probe 12 in. long, Whitney Instrument Co.
152	9	Half Shielded Temperature Probes 12 in. long, Whitney Instrument Co.
153	4	Half Shielded Temperature Probes 16 in. long, Whitney Instrument Co.

MANOMETERS

Item No.	Quantity	
154	1	1/2 in. inclined draft gauge, Friez
155	2	1/2 in. inclined draft gauge, Meriam, No. G P-1
156	2	1 in. inclined draft gauge, Ellison No. 11440
157	2	1 1/2 in. inclined draft gauge, Ellison
158	2	2 in. inclined draft gauge, Ellison No. 11450
159	2	3 in. inclined draft gauge, Ellison No. 11470
160	1	1 in. Marine Type Hays No. 1-BPD-1, Range 0-1 in.
161	1	Harrison Micromanometer
162	1	Micromanometer (Whalen Type)
163	1	30 in. U type Manometer, Meriam "Clean-out"

VARIABLE AREA FLOW METERS

Item No.	Quantity	
164	2	Fisher-Porter Flowrator Tubes No. 08-150
165	2	Fisher-Porter Flowrator Tubes No. 01N-150-A
166	2	Fisher-Porter Flowrator Tubes No. B4N-250-A-L
167	2	Schutte & Koerting 3/4-in. Rotameters, Fig. 1891

ELECTRIC METERS

Item	No.	Mfr.	Model	Scale	
168	1	Weston	433	30/15	A.C. Voltmeter
169	1	Weston	433	150/75	A.C. Voltmeter
170	1	Weston	433	300/150	A.C. Voltmeter
171	1	Hickok	47	300/150	A.C. Voltmeter
172	1	Weston	430	70/7.5/3	D.C. Voltmeter
173	1	Weston	433	1	A.C. Ammeter
174	1	Weston	433	5/2.5	A.C. Ammeter
175	1	Weston	433	10/5	A.C. Ammeter
176	1	Weston	433	50/20/5	A.C. Ammeter
177	8	Hickok	47M	5	A.C. Ammeter
178	1	Hickok	47M	25	A.C. Ammeter
179	1	Weston	433	75/15	A.C. Milliammeter
180	1	Weston	433	300/150	A.C. Milliammeter
181	1	Weston	433	500/250	A.C. Milliammeter
182	1	Weston	430	1500/150/15	D.C. Milliammeter
183	1	Weston	931	3000/300/30	D.C. Milliammeter
184	1	Simpson	27	1	D.C. Milliammeter
185	1	Elec. Inst.		5	D.C. Ammeter
185	1	Weston	45	15/115	D.C. Ammeter
187	1	Weston	310	100	A.C. & D.C. WM
188	1	Weston	310	500	A.C. & D.C. WM
189	1	Hickok	19W4	300/150	A.C. & D.C. WM
190	1	Hickok	450		Volt-Ohm-Milliammeter
191	1	RCA	WV-77E		Volt Ohmyst vacuum tube VM
192	1	Pyr. Inst.	300		Amprobe snap-around VAM
193	1	Duncan			Rotating Standard Portable Test Watt-hr meter
194	2	Weston	461		Type 1 5 volt-amp Capacity Current Transformers

POTENTIOMETERS

Item	No.	
195	2	High Precision Potentiometers, Rubicon Type B, Cat. No. 2780
196	1	Portable Precision Potentiometer, Leeds & Northrup, Cat. No. 8662
197	1	Portable Precision Potentiometer, Tubicon, Cat. No. 2702
198	1	Single Range Portable Potentiometer, Leeds & Northrup, Cat. No. 8656-B
199	1	Vernier Precision Potentiometer Tinsley
200	1	Ionization Potentiometer Tinsley
201	1	Electronik precision indicating Potentiometer, Brown Model No. 156x63V12 complete with parts to provide ranges of 0-5, 0-10 and 0-50 mv
202	1	Precision indicating Potentiometer, Brown Electronik Model No. 156x15V, with 0-5 mv scale
203	2	12 point recording Potentiometers, Brown Electronik Model Y153x60P12-X-61(V) Scale—30-230F
204	1	12 point recording Potentiometer Brown Electronik Model Y153x60P12-X-61(V) Scale 0-10 mv
205	1	Voltage Recorder, Foxboro EMF Dynalog Model 9350N, Range 0-5 mv

GALVANOMETERS

Item	No.	
206	1	Rubicon dc Spotlight Galvanometer, Cat. No. 3400-H
207	1	Rubicon dc Spotlight Galvanometer, Cat. No. 3417
208	3	Leeds & Northrup dc Moving-Coil, Reflecting Galvanometers No. 2500-a, Type R.
209	1	Tinsley Reflecting Type Galvanometer
210	1	Weston Galvanometer, Model 1440
211	1	Leeds & Northrup Galvanometer, Cat. No. 2430 A
212	3	Galvanometer Lamps

RADIATION INSTRUMENTATION

Item	No.	
213	1	Pyriheliometer, Smithsonian, silver disc type
214	1	Radiometer, Kipp & Zonen, Type E5, No. 2766
215	2	Phototube Units, Fisher Scientific Co. No. 13-991-5
216	2	Electronic Relays, Fisher-Serfass, No. 13-991
217	1	Lamp Unit, Fisher Scientific Co. No. 13-991-10
218	1	Photronic Cell, Weston No. 594
219	2	Photovoltaic Cells, General Electric, Model 8 PV1
220	1	Standard Radiation Source, National Bureau of Standards Lamp

SOUND INSTRUMENTATION

Item	No.	
221	1	Ilg Mfg. Co. Standard Sound Source
222	1	Random-Noise Generator, General Radio, Type 1390-A
223	1	Sound Level Meter, General Radio, Type 1551-A
224	1	Sound Analyzer, Scott, Type 420-A
225	1	Sound Level Meter, Electrical Research Products, No. RA-245
226	1	Sound Filter Set, Electrical Research Products, No. RA 243, Series 7

SCALES AND BALANCES

Item	No.	
227	1	Analytical Balance, Ainsworth Type DL., Fisher Scientific Co. No. 1-931
228	1	Set Class S Balance Weights, 5 mg to 100 g, Fisher No. 2-214
229	1	Set Class C Balance Weights 1 cg to 200 g, Chemical Rubber Co. No. 665-S
230	1	Set Class C Balance Weights, 1 cg to 500 g, Chemical Rubber Co. No. 665-S
231	2	Triple Beam Balances, Central Scientific No. 3620, Cap. 2110 gram
232	1	13 x 20-in. Platform Scales, Fairbanks Morse, Beam 3 lb x ¼ oz, Cap. 300 lb
233	1	16 x 25-in. Platform Scales, Fairbanks Morse, Code 1204, Beam 2 lb x 0.01 lb, Cap. 400 lb
234	1	17 x 27-in. Platform Scales, Fairbanks Morse, Code 1124, Beam 50 lb x ¼ lb, Cap. 1000 lb

MISCELLANEOUS INSTRUMENTS

Item	No.	
235	1	Hygro-Thermograph, Friez, Model 594
236	1	Selsyn Wind System, Friez, consisting of 1—No. 1020 Direction Transmitter 1—No. 1030 Velocity Transmitter 1—No. 377-18 Support 1—No. 403 Indicator Panel
237	1	Mercurial Barometer, Fisher, No. 2-380
238	1	Precision Aneroid Barometer, Bendix-Friez, Model 999A with Leather Carrying Case
239	1	Set of Hydrometers, Precision Thermometer & Inst. Co., Cat. No. 807
240	1	Westphal Specific Gravity Balance, Fisher No. 2-150
241	1	Recording Thermometer, 10 to 100F, Trerice No. 3050
242	2	Recording Thermometers, 0 to 100F, Taylor Instrument Co.
243	1	Temp. & Humidity Recorder, 0 to 100F, Bristol Model 4069
244	1	Skin & Rectal Temp. Indicator, Alnor, Type 2900
245	1	Metabolism Apparatus, Benedict-Roth
246	1	Blood Pressure Apparatus, Baumanometer, Kompak Model
247	4	Stop Watches
248	1	Microscope, Spencer No. CT 32357, with stage micrometer
249	1	Microprojector, Mine Safety Appliance Co., No. CT45823 Dust Vue, complete
250	15	16 in. Long Sliding Contact Rheostats, Misc. Capacities
251	3	Decade Resistance Boxes, General Radio Co., Type 602F
252	1	Wheatstone Bridge, Leeds & Northrup No. 5305 Test Set
253	1	D.C. Microvolt Amplifier, Leeds & Northrup No. 9835A
254	2	Portable Orsats—Hays Improved Gas Analyzers
255	1	Orsat, Burrell Tech. Supply Co. Model VRE, No. A39-537
256	3	Magnetic Counters, Veeder-Root No. UD, 6 volt dc
257	1	Magnetic Counter, Veeder-Root No. US, 110 volt—60 cycle
258	1	Mechanical Counter, Durant Productimeter, Model 6-C-3-S
259	2	Micro Pipet-Burets, Gilmont Combination G 15395 C

Meetings ahead

May 9-12 – Mechanical Contractor's Association of America, 72nd Annual Convention, Miami Beach, Fla.

May 16-18 – Building Research Institute, Spring Conferences, Washington, D. C.

May 21-24–Industrial Heating Equipment Association, Hot Springs, Va.

May 22-25–Design Engineering Show, Detroit, Mich.

June 11-14–American Society of Mechanical Engineers, Summer Annual Meeting, Los Angeles, Calif.

June 12-15–National District Heating Association, 52nd Annual Meeting, Portsmouth, N. H.

June 12-15 – Institute of Boiler and Radiator Manufacturers, Annual Meeting, Absecon, N. J.

June 25-30 – American Society for Testing Materials, Annual Meeting, Atlantic City, N. J.

June 26-28 – American Society of Heating, Refrigerating and Air Conditioning Engineers, 68th Annual Meeting, Denver, Colo.

October 2-4 – American Gas Association, Annual Convention, Dallas, Texas.

October 23-27–National Metal Exposition, Detroit, Mich.

October 31-November 2–Fourth Canadian Refrigeration and Air Conditioning Show, Toronto, Ont.

November 5-7–National Frozen Food Association, National Convention and Exposition, Bal Harbour, Fla.

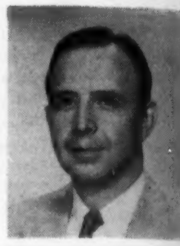
November 6-10 – National Warm Air Heating and Air Conditioning Association, 48th Annual Convention, Chicago, Ill.

November 12-15 – Air Conditioning and Refrigeration Institute, Annual Meeting, Hot Springs, Va.

November 13-17–National Electrical Manufacturers Association, Annual Meeting, Atlantic City, N. J.

News of ASHRAE members

New jobs



W. Donald Geiser has been promoted to Vice President in charge of production for Baltimore Aircoil Company, Inc. Production Manager for the past year and a half, he was Chief Design Engineer for two years prior to that. A graduate of Wayne University, he has more than 15 years experience in various engineering development and production positions.

Sydney Anderson has been appointed Vice President in charge of West Coast operations for Typhoon Air Conditioning Div, Hupp Corporation. He will supervise activities of the organization in California, Arizona and Nevada. Prior to joining Typhoon, he was Los Angeles Branch Manager for Chrysler Corporation Airtemp Div. In addition to ASHRAE, he is a member of the American Management Association and past-Chairman of the Unitary Section of the Air-Conditioning and Refrigeration Institute.

Lawrence E. Jennings, Jr., advances to the post of Manager-Refrigeration, Bohn Aluminum & Brass Corporation, Danville Div. Following graduation from Trinity College, he worked as a sales engineer for several years and as Chief Engineer for a manufacturer of refrigeration and air conditioning equipment. He joined Bohn in 1960.

Donald A. McKay has been named Sales Manager and **John J. McKee** is now Estimating Manager of Limbach Company's Boston office. Mr. McKay, a 1952 graduate of Harvard University, joined Limbach in 1960, following five years as a sales engineer with Charles P. Blouin, Inc. Mr. McKee was graduated from Carnegie Institute of Technology in 1955.

Joseph D. Loveley has been appointed Coordinator of Manufacturing and Engineering for Copeland Refrigeration Corporation. For the past three years he has been Assistant Chief Engineer in Chrysler Corporation's Central Engineering Department, in charge of vehicle air conditioning, heating and ventilating. Previously he was Vice President in charge of Engineering for Chrysler Airtemp. He is a 1934 graduate of the University of Detroit.

Ray Long has been named Branch Manager of the Detroit Sales Office of Ilg Electric Ventilating Company. Prior to this transfer, he was Branch Manager of the Baltimore office for six years.

E. W. Ervasti has been appointed to the newly formed post of Assistant Director of Marketing for Wolverine Tube Div, Calumet & Hecla, Inc. Beginning his career with Wolverine in 1934, he since has worked in operational and marketing capacities. In 1957 he was advanced from the post of Manager of Industrial Sales for the American Div to General Sales Manager for Canada.

Frederick M. Long is now Assistant Sales Manager for Baltimore Aircoil Company, Inc., following four years as a sales engineer for Trane Company, specializing in industrial sales.

George J. Finck has been appointed Product Manager, Refrigeration, of Dunham-Bush, Inc. An alumnus of Grinnell College, he has been with the company for more than 13 years and most recently was New York District Sales Manager. Past-Chairman of New York Section of the former ASRE, he is currently Vice Chairman of the ASHRAE Admissions and Advancements Committee and a member of the General and Administrative Coordinating Committee. In addition to ASHRAE, he is a member of Refrigeration Service Engineers Society.

William O. Huebner, formerly with Anemostat Corporation of America for more than twenty years, will set up his own office as consultant, writer and editor. Having served as Anemostat's Technical Editor, he also wrote every issue of Aspiration. He received his degree in mechanical engineering from the Technical University, Berlin, Germany.

Kenneth A. Merz is now Director of Engineering of the Air Moving Div of Torrington Manufacturing Company, in which post he will be responsible for setting overall standards for air moving engineering. With Torrington in various engineering capacities for more than 12 years, he has for the last year been Engineering Manager of the Torrington, Conn., Air Impeller Div. Previously he was Engineer in Charge of Product Development, Assistant Chief Engineer and Production Engineering Manager. He is an alumnus of Yale University.

Robert J. Evans, who has served as a design engineer with a number of manufacturers of residential air conditioning, domestic refrigeration and mobile refrigeration and air conditioning for the past 14 years, has joined the staff of the Air-Conditioning and Refrigeration Institute as Chief Engineer. He received his B.S. degree in mechanical engineering from the Illinois Institute of Technology.

Norris M. Blanchard, head of Norris M. Blanchard Company, has been appointed manufacturer's representative for the Heating and Air Conditioning Div of Stewart-Warner Company. He founded his organization in 1947. Newly associated with him is **Kenneth R. Magarrell**, past-President of ASHRAE Nebraska Chapter. His prior connections were with Baker Manufacturing Company, from 1931 to 1944, and the Gallaher Company.

Honors and Recognitions

H. C. Diehl, Fellow ASHRAE and Honorary Life Member of the Refrigeration Research Foundation, has been chosen to receive the Nicholas Appert Award of the Institute of Food Technologists, consisting of a gold medal and a prize of $1000. This award is made each year to a man "who has distinguished himself by outstanding leadership and research contributions in the field of food science and technology." Dr. Diehl received the honorary degree of Doctor of Science from the University of Rhode Island in 1957, was national President of the Institute of Food Technologists in 1948, is a member of the International Institute of Refrigeration and Secretary of its Commission IV as well as a member of the IIR U. S. National Committee.

Necrology

John H. Spence, 66, Education Director of Refrigeration Service Engineers Society and former National Service Manager for Hussmann Refrigeration, Inc., died March 21. Acting Education Director since 1957, he became full-time Director in 1960, following his retirement from Hussmann. At Hussmann, he was responsible largely for development of the company's present field service organization, factory warranty service and field liaison with the engineering department. His educational activities with RSES, especially the annual programs and increased size of the service manual, have earned him much recognition by the industry.

Jack Holmes Waggoner, Technical Assistant for Pittsburgh Plate Glass Company, died on February 21. He had, from 1937-43, served on the ASHVE committees on Air Filtration Devices and Thermal Insulation.

George J. Redmond, Jr., deceased, was 44. He had been a member of Kansas City Chapter.

Clarence D. Berkeley died Feb. 27 at the age of 52. His business affiliation had been with General Electric Company.

Others are saying—

ammonia absorption machines . . . may be designed more exactly than previously, justifying an extension of their use, especially in the chemical industry where inexpensive steam is available. Offered in this paper is an economic evaluation of this equipment, based on new data and on normal chemical engineering procedure. Design calculations were carried out for machines operating at evaporator temperatures of zero, —30 and —60 F and at capacities of 1,000,000 and 5,000,000 Btu/hr in each case. *Canadian Refrigeration and Air Conditioning, February 1961, p 24.*

high pressure air . . . was used to meet strict requirements in air conditioning the 30-year old Federal Building in Newark, N. J. Problems presented were: no presently used floor space to be taken for the system, conditioning in courtrooms and judges' chambers to be at a low level of sound intensity, system for the FBI areas to be separate and work to proceed with least possible disturbance to normal activities. Combined to solve these problems were: use of an existing steam-return heating system, a new type of air induction terminal outlets to replace cast iron radiators and high pressure cooled air delivered by 25 remotely located air handling units of various capacities. *Refrigeration & Air Conditioning Business, April 1961, p 50.*

high intensity infra-red heaters . . . for comfort heating of large volume industrial and commercial spaces are discussed in a comprehensive report. Types of units available, spot heating of scattered work stations, providing comfort in limited areas, design for total space heating and relation of comfort and heat input are among items covered. *Heating, Piping & Air Condiitoning, January 1961, p 229.*

CROSS FLOW COOLING TOWERS ANALYZED GRAPHICALLY PAGE 53

What ASHRAE Regions and Chapters are doing

Problem areas covered at meetings include vibration and noise control, air distribution, environment, water treatment and air pollution. Other Chapters toured such varied sites as a power plant, a bakery and a manufacturing plant.

NORTHERN CONNECTICUT . . . Present goal of the gas industry, stated W. Roger Sarno of the American Gas Association, speaking at the February 9 meeting, is to use fuel cells and thermo-ionic means of generating thermo-electric power. This would enable basic gas-powered equipment to be self sufficient and not require supplemental electricity to drive fans, pumps and such.

"Thermoelectric cooling is presently competitive in the range of 160 Btu/hr", stated March speaker Robert S. Lackey of Westinghouse Electric Corporation. Included in his presentation were a description and discussion of the Peltier and Seebeck effects and what presently is being done to utilize this mechanism.

who's doing what . . . R. Miller, S. Badula and T. Carey have been selected as tellers for the ballot count in election of officers for the 1961-62 season.

MINNESOTA . . . Professor Warren Harris of the Institute of Boiler and Radiator Manufacturers Committee, University of Illinois, presented a discussion at the January meeting on the IBR Research Home located at the University. Among items covered was redesign of a hydronics system in the Home. When water temperature was increased from 170 to 215 F, heating surface requirements were reduced 40%, while operational costs were increased but 5%. Piping sizes were reduced substantially, as were installation costs.

OREGON . . . Presented at the March 16 meeting were resumes of the various technical sessions, symposiums and forums of the Chicago meeting. Earl Constant covered the ventilation symposium and the First Technical Session, Omer Jacobsen the Second and Wayne Gibson the Third; Donald L. Benz reviewed a symposium on air movement and humidity control; Kenneth R. Murhard reported on medium temperature water systems; J. Donald Kroeker covered a symposium on the Society research program and a forum on psychrometric charts; and Chester Jarrett discussed a forum on environments for learning and a field trip to the Dresden Nuclear Power Station. Owen Matthews then reviewed the Heating and Air-Conditioning Exposition, followed by Bert Farnes, who covered National Committees and their actions.

who's doing what . . . Elected to office for the 1960-61 season are: Earl Constant, President; James L. Waymire, Vice President; Chester W. Timmer, Secretary; William Maxwell, Treasurer; and Omer Jacobsen and K. Gilbert Stahl, Board of Governors. Donald H. Markman is Chairman of the local UEC Building Fund Drive.

CINCINNATI . . . John P. Soule of American-Standard Industrial Div spoke on "The Air Conditioning Market" at the March 7 meeting. A question and answer period followed.

James Snyder reviewed comments from the National By-Laws Committee, pointing out the sections which require revision to comply with recommendations of the Committee.

who's doing what . . . J. Stockwell has been appointed Chairman of the School Environment Committee, members of which are Roger Anderson and Hal K. Jennings.

WISCONSIN . . . Main speaker at the February 20 meeting was Russ Brown of Arkla Air Conditioning Corporation, whose topic was "Absorption Refrigeration". "Heat Pumps" were covered by Lee Miles of Mueller Climatrol.

who's doing what . . . E. Zieve reported for the Membership Committee that seven new applications have been filed.

ATLANTA . . . Beginning the year was an address on the "Technology of Building Automation Control Systems," presented at the January meeting by Jack Marshall of Minneapolis-Honeywell Regulator Company. Featured in his talk were slides showing various types of control systems.

In February, Lt. Colonel Sid F. Spear, Director of Information at the Air Force Missile Center,

CHAPTER MEETING DATES

	May	June		May	June		May	June
Alamo	—	—	Central Pennsylvania	10	—	Illinois	8	—
Arkansas	16	20	Cincinnati	—	—	Illinois-Iowa	15	—
Atlanta	—	—	Cleveland	8	—	Inland Empire	8	11
Austin	18	15	Columbus	15	19	Iowa	8	11
Baltimore	4	—	Dallas	15	19	Jacksonville	—	—
Baton Rouge	16	—	Dayton	—	—	Johnstown	9	—
Boston	—	—	El Paso	15	19	Kansas City	1	—
British Columbia	10	—	Evansville	2	3	La Ville de Quebec	9	—
Central Arizona	1	2	Florida West Coast	—	—	Long Island	—	—
Central Indiana	9	—	Fort Worth	17	21	Louisville	8	—
Central Michigan	9	13	Golden Gate	—	—	Manitoba	25	—
Central New York	17	—	Hampton Roads	2	6	Memphis	15	19
Central Oklahoma	8	—	Houston	19	16	Michigan	—	—

Patrick Air Force Base, Fla., spoke and showed a film on missile launchings and test facilities in the Florida Keys.

Following a March program on "Sound Noises" by J. Barrie Graham, Research Director of Buffalo Forge Company, this Chapter is planning a tour of the facilities of the Georgia Power Company's new 22-story building.

SAN JOAQUIN . . . Beginning the March 21 program was a short film on electrical problems in air conditioning systems. After the film, Don Jensen of Westinghouse Electric Corporation discussed types of motors, motor control equipment and speed controllers.

SHREVEPORT . . . At the March 16 meeting, Richard Johnson, Chapter Vice President and Chairman of the Regional Meeting Committee, reported on proposed programs for the meeting, to be held April 20-22.

NEBRASKA . . . W. Bell, R. Kulicek, A. Blue and C. Goth discussed "Electric Heating" at the March 14 meeting. B. G. Peterson was Moderator.

PITTSBURGH . . . Basic theories of vibrations, frequencies and effectiveness of damping vibrations were explained at the February 20 meeting by Jack Harris of Korfund Company, speaking on "Vibration, Shock and Noise Control for Air Conditioning Equipment." Using slides, he showed application values of cork, rubber and springs. Various types of vibration mounting units were illustrated and slides of actual installations indicated uses of mountings for fans, pumps, cooling towers and compressors.
who's doing what . . . Art Nass, Jr., Chairman of the local UEC Fund Drive, reported on progress of this Chapter.

LOUISVILLE . . . Featured at the February Technical Seminar were presentations on evaporating refrigerants by Merl Baker of the Kentucky Research Foundation and Will K. Brown of the University of Kentucky.

Dr. Baker started the session with a review of the mechanism of evaporating refrigerants, describing four different processes by which evaporation takes place. Under conditions of low temperature difference between wall and refrigerant, evaporation is basically a free convection boiling process with vapor forming at the liquid vapor interface. As the temperature difference is increased, bubbles are formed at the tube wall and nucleate boiling occurs. Under these conditions the wall is wetted by liquid refrigerant. Increasing the temperature difference further, the tube wall becomes wetted partially and then is covered with a film of vapor. This condition is designated film boiling.

Because of the differences that occur in the heat transfer processes for each condition of boiling, basic information is required for the design of evaporators. Results of studies of this type conducted at the University of Kentucky were presented by speaker Brown. Tested was a cylindrical heater which can be rotated in a concentric bath of refrigerant. By varying the input to the heater and the rotational speed, the heat transfer rates can be varied through the complete range of processes. While the tests are not finished, the researchers contend that they have a method by which they can acquire basic data of interest to designers of refrigeration equipment.

"Environmental Problems in Space Travel" was the subject of discussion at the regular February meeting, when Dr. Karl O. Lange of Lange Instrument Company was guest speaker. Temperature, pressure, radiation, abrasion by micrometeorites and lack of gravity can be overcome, he contended. The heat of re-entry is dissipated in a form of sweating; a metal or ceramic coating outside the ship is caused to melt away to prevent penetration of the heat. Also mentioned were problems due to acceleration, deceleration and vibration.

WESTERN MICHIGAN . . . Problems of air pollution and what is being done in legislation were detailed by Bernard D. Bloomfield, Assistant Director, Div of Occupational Health, Michigan Department of Health, guest speaker at the March 13 meeting.
who's doing what . . . Results of the election of officers for the 1960-61 season are: G. L. Jepson, President; D. A. Rackliffe, First Vice President; William Wessels, Second Vice President; L. H. Hinkel, Secretary; C. W. DeKorte, Treasurer; and S. R. Curtis, R. J. Waalkes and J. F. Belton, Board of Governors. · R. J. Waalkes and D. J. Renwick have been nominated Delegate and Alternate to the Chapters Regional Meeting.

UTAH . . . Held in conjunction with Engineers' Week at the University of Utah, the February 17 meeting of this and other participating Chapters and groups was attended by more than 800 engineers. In addition, the meeting was televised on a closed circuit

	May	June
Middle Tennessee	9	13
Minnesota	8	—
Mississippi	22	26
Mobile	22	26
Montreal	15	—
National Capital	10	—
Nebraska	9	13
New Mexico	—	—
New Orleans	16	—
New York	23	—
Niagara Frontier	1	5
Niagara Peninsula	2	—
North Alabama	—	—
North Jersey	—	—
North Piedmont	12	—
Northeastern New York	15	—
Northeastern Oklahoma	—	—
North Alberta	—	—
Northern Connecticut	18	—
Northern Ohio	—	—
Ontario	1	—
Oregon	11	15
Ottawa Valley	—	—
Panama & Canal Zone	—	—
Philadelphia	25	—
Pittsburgh	15	—
Puget Sound	9	—
Rhode Island	10	—
Richmond	—	—
Rochester	17	—
Rocky Mountain	—	—
Sacramento Valley	—	—
St. Louis	15	—
San Diego	9	13
San Joaquin	—	—
Savannah	—	—
Shreveport	18	15
South Carolina	15	—
South Florida	9	—
South Piedmont	—	—
Southern Alberta	16	20
Southern California	8	15
Southern Connecticut	11	8
Toledo	8	—
Tucson	9	—
Utah	19	—
West Texas	26	—
Western Massachusetts	18	16
Western Michigan	—	—
Wichita	15	—
Wisconsin	15	—

to Kingsbury Hall on the University Campus, where an additional 2000 high school and University students could hear the guest speaker, Dr. Wernher Von Braun, Director of the George C. Marshall Space Flight Center at Huntsville, Ala.

Accomplishments of many scientists were cited by Dr. Von Braun, who also outlined advances in rocketry since launching of the first Sputnik three years ago. Mentioned were weather observation satellites, satellite recovery and discovery of the Van Allen radiation belt. Concluding the program was a film on the Saturn rocket.

PHILADELPHIA . . . Dr. James R. Anderson of the Applied Research Department, Defense Electric Products Div, RCA, speaking on "Thermoelectric Refrigeration" at the March 9 meeting, stated that the present cost of thermoelectric materials is approximately $20,000 per ton of cooling. Power operating costs are about three times that of conventional refrigeration at present, he added. These factors make such refrigeration prohibitive except in cases where the advantages outweigh costs. According to Dr. Anderson, advantages are: costs become more competitive in small size units, excellent reliability because of lack of moving parts, silent operation and suitability to modular construction.

At the pre-dinner educational session, Ellis I. Currey, Sr., Sales Engineer for American-Standard Industrial Div, spoke on "Air Flow Measurement". Problems discussed involved measurement of fan air deliveries in the field.

RHODE ISLAND . . . Nuclear energy, its origins and applications, was covered at the March 8 meeting by Paul Moffatt, Director of Quality Control, Metals and Controls Div, Texas Instruments, Inc.
who's doing what . . . Elected to serve for the 1961-62 season are: Clifford H. Dow, President; Robert E. Wilkinson, Sr., Vice President; Daniel J. Kiely, Jr., Secretary; John K. Maclean, Treasurer; and Frank Sarra and Leo McPherson, Board of Governors.

BOSTON . . . MIT Research Associate C. B. Engebretson spoke at the March 21 meeting on a test house heated and cooled with solar energy. Slides were shown of the design of the house, construction of the solar heat collector, method of storing the heat in a tank containing 1500 gal of water and distribution of the water to the house. Tests during one year showed that 46% of the heat required for the house could be supplied by solar energy.
who's doing what . . . Tentatively proposed for the Nominating Committee are John Bonner, Richard Forbes, William Warner, Arthur Hare and Wilder Parks.

SACRAMENTO VALLEY . . . At the March 1 meeting Maurice J. Wilson of Carrier Corporation commented briefly on air conditioning in schools. Water treatment for cooling systems was the topic of the evening's main speaker, Frederick S. Hodgdon of Metropolitan Refining Company. Various categories of water treatment problems were illustrated by a film, "The Nature of Corrosion—Anode and Cathode Process."
who's doing what . . . Vern Thornberg, Chairman of the Research Committee, reported that collection of design weather data is to begin. T. Andrews and L. Stecher are to be Chapter representatives to the Region X meeting in Phoenix on April 28 and 29.

MIDDLE TENNESSEE . . . John Tudor moderated a panel discussion on "Base Bid Specifications" which was held at the February 21 meeting. Comprising the panel were Don Orr, Andrew Reid, Ray Gardner and Lester Smith.

Speaking at the March 14 meeting was W. W. Caines of Carrier Air Conditioning Company, M. S. O. Div.

SOUTHERN CALIFORNIA . . . ASHRAE Past-President Daniel D. Wile was guest speaker at the March 13 meeting. Discussing "Modern Trends in Commercial and Industrial Refrigeration," he gave a capsule history of early refrigeration methods and traced the development of four distinct components: heat transfer design, control devices, application engineering and defrosting methods. Heat transfer surfaces have changed from bare pipe coils dependent on gravity for air flow to finned surface coils in ceiling-suspended forced air unit coolers. Modern control devices make possible much closer control of temperature, humidity and system operating conditions. Speaker Wile compared pumped hot gas vapor systems of defrost with water and electric defrosting. In discussing evaporative condenser design, he showed illustrations of systems using water, air-cooled and closed air circuit types. In closing, he commented on brine, ethylene glycol and flooded ammonia refrigerant systems. Throughout his discussion, Past-President Wile illustrated his talk with slides and descriptions of refrigeration plants designed recently, showing the application and operating advantages over earlier systems.

Regional Director T. J. White was present at the March meeting to report on matters discussed at the national meeting in Chicago concerning legislation activities by chapters, especially Codes and Standards Committee work. He also announced the program for the Regional Meeting in Phoenix, April 28 and 29.

COLUMBUS . . . Meeting jointly with the Columbus Technical Council on February 23, members heard an address by George F. Taubeneck, Publisher of Air Conditioning, Heating and Refrigeration News.

Coffee Talk Speaker at the March 20 meeting was Edward Wuesthoff, Application Engineer for Room Air Conditioners at Westinghouse Electric Corporation; his subject was "Through-the-Wall Air Conditioners."

"Refrigerant Systems Drying" was covered by the evening's main speaker, Stephen Balough, Chief Engineer, Kenmore Machine Products, Inc.

NEW YORK . . . With the national interest today centered on the probing of space, refrigeration engineers are being challenged by the need for systems which can duplicate, in environmental test chambers, tem-

peratures and other conditions encountered in outer space. Not only are the conditions themselves relatively difficult to achieve, but the system delivering these conditions must be able to do so in rapidly changing sequence. Reviewing the design of and the equipment selection procedure for multistage refrigeration systems for low temperature applications, March speaker E. Zuckerman of Arthur E. Magher Company described some of the systems now in use.

Featured at the third of the Spring series of Technical Seminars was a discussion of "Fan System Application" by George Norman of Buffalo Forge Company and Harold Brush of Westinghouse Electric Corporation.

TUCSON . . . Missile development was covered by Major Melvin Dart of the U. S. Air Force, guest speaker at the March meeting.
who's doing what . . . Candidates for Chapter offices are: President, J. H. O'Hair and D. D. Shipley; Vice President, J. P. Jones and D. Means; Secretary, A. E. Hamilton and J. Kemp; Treasurer, B. M. Dehlinger, H. Dinwiddie and D. Miller; Board of Governors, F. Blackmore, D. Haskett, G. Smith and D. Sterrett.

OREGON . . . Nominated for the 1961-62 term are: Earl Constant, President; James L. Waymire, Vice President; Chester W. Timmer, Secretary; William Maxwell, Treasurer; and K. Gilbert Stahl, Board of Governors.

ROCHESTER . . . March speaker J. R. Fortier of Westinghouse Electric Corporation gave a talk on "Thermoelectric Refrigeration." In addition to cooling, he touched on thermoelectric power generation.
who's doing what . . . Henry Dyminski reported on the Semiannual Meeting in Chicago. Lawton Engelhart announced that a chapter mailing would be started for the UEC Fund Drive.

SOUTHERN ALBERTA . . . At the February 21 meeting J. Orr of Honeywell Controls talked on "Control of High Velocity, Double Duct Systems", J. Patterson of Johnson Controls covered "Control of High Velocity Induction Systems" and J. Glennon of Powers Controls discussed "Fan Coil Systems".
who's doing what . . . W. Jack is Chairman of the Nominating Committee.

JOHNSTOWN . . . February speaker was W. R. Eaton, Manager of Quality Control and Test Operation for General Electric Company's Missile and Space Vehicle Department. Covered was "Private Enterprise in the Space Industry."

IOWA . . . Evolution of gas absorption refrigeration equipment was traced by March speaker Donald W. Munson of Trane Company, beginning with the origin of this type of unit around 1760. Prior to 1935, most of the absorption machines were of the ammonia-water type, and were limited principally to special process applications because of the difficulty involved in recovering ammonia from the water, due to the close boiling points of the two. Many combinations of refrigerant and water were examined before acceptance of today's lithium bromide-water type.

Pointed out was the fact that a lithium bromide refrigeration system is inherently difficult to handle for three basic reasons: it is corrosive when air is allowed into the system, it is a salt and tends to revert to its crystalline form and the boiling points of water and lithium bromide are close enough together to present a problem in refrigerant reclamation.

In the 1940s larger systems were designed, incorporating pumps, and early in 1950 lithium bromide machines were designed with a second stage of heat exchangers. This permitted the system to be designed so that low pressures of 1-1/100 of an atmosphere could be confined to a machine. Today's factory packaged machines, using new types of pump seals, eliminating gasketed joints and using an all-welded technique are cited as having eliminated the problem of air entrance to the system.

KANSAS CITY . . . R. T. Sheridan, Executive Vice President and Sales Manager of Weil Pump Company; A. J. Solari, Chief Liaison Engineer, Crane Packing Company; Ray Perkins of Scott & Kinney; and Mr. Wolfe of Western Chemical Company were members of a panel which discussed pumps and pump seals at the March 6 meeting. Brought out was the point that chromates in proportions normally used in water treatment are not injurious to the seal.
who's doing what . . . Proposals of the Nominating Committee included William Scurlock, President; Edgar M. Hopkins, First Vice President; Ernest Engelhardt, Second Vice President; Secretary, Paul Moreno; Treasurer, Robert Kilker; and Board of Governors, A. Sneller, R. Perkins, G. Hart and H. Burkart.

WESTERN MASSACHUSETTS . . . February speaker Edward Tarnoff of Tarnoff Associates spoke on natural gas driven compressor units of the type used in air conditioning. Covered were operating characteristics, performances and costs.

BRITISH COLUMBIA . . . A new feature of meetings is a talk on a "Topic of the Month" which, at the February 15 meeting, was presented by Richard Perry. Speaking on "Air Conditioning Design of the Van Tel Studios", he gave figures on energy-cost relationship and on savings effected by eliminating a boiler. By means of a flow diagram he showed the system layout for heating and cooling.

Main speaker of the evening was A. G. Kaneen, Chief Inspector, Gas Inspection Div, Department of Public Works. He indicated problems resulting from inadequately sized boiler and furnace rooms, pointing out the need for clearance space around equipment for maintenance and inspection. Also mentioned were factors affecting the supply of air to gas burners. Discussed were draft, venting, chimneys and design for breach induced draft fans between boiler and stack.
who's doing what . . . Proposed by the Nominating Committee as officers for the 1961-62 season are: President, David Leaney; Vice President, Robert Hole; Secretary, William Baker; Treasurer, James

Phillipson; and Board of Governors, J. M. Bird, Robert Butler, Paul Daum, Frank Dwyer, Robert Millar, William Swarbrick and R. Racine.

ST. LOUIS . . . Changes in the St. Louis Building Code, as cited by Charles J. R. McClure at the March meeting, include: with natural ventilation, window area is to be 1/10 of the floor area; kitchens must have three cfm per person per ten sq ft of floor space; assembly rooms must have ten cfm per person; work rooms must not be more than 10 F above outside temperatures; and stationary engineers will be required for small boilers operating at more than 15 psi and 250 F.

who's doing what . . . Officer candidates for the 1961-62 season are: F. G. Meyers, President; J. B. Killebrew, First Vice President; R. B. Tilney, Second Vice President; H. R. Halt, Recording Secretary; C. F. Barmeier, Corresponding Secretary; C. W. Baker, Treasurer; K. Williams, J. F. Cuba and C. Hartung, Board of Governors. George Myers has been named Convention Chairman and Bruce Evans Vice Chairman.

CENTRAL PENNSYLVANIA . . . Outlining factors affecting air distribution during cooling and heating, with emphasis on the effects of floor, baseboard, sidewall and ceiling outlets, Harold E. Straub, Chief Research Engineer of Titus Manufacturing Corporation, spoke at the March 8 meetings. Slides showed the results of a number of experiments and tests conducted at the University of Illinois while speaker Straub was working on an air distribution program there.

CLEVELAND . . . Controls were the subject of a panel discussion held at the March 13 meeting. Basic control terminology was outlined by John F. Krohn of Minneapolis-Honeywell Regulator Company, placing special emphasis on a type of reset required in certain control applications to eliminate offset. W. A. Wohl of Johnson Service Company covered the importance of proper location of temperature sensing elements. Representing Barber-Colman Company, Ben Healy discussed sizing control valves. James Potter of Powers Regulator Company concluded with a talk on damper sizing, stating that a higher pressure drop across automatic control dampers is required if they are expected to modulate.

ILLINOIS . . . Panel members discussing "Who is Responsible for Freeze-Ups?" at the March 13 meeting were M. G. Kendrick, H. W. Alyea, R. K. Lee and S. I. Rottmayer. Emphasized was the need for provision of continuous circulation in water coils, positive drainage and even air distribution over the face of the coils.

who's doing what . . . Regional Director L. K. Warrick spoke on the research program of the Society.

HOUSTON . . . "Refrigeration as Applied to the Dairy Industry" was discussed at the February 17 meeting by Ray Ferguson of Carnation Company. Rates of bacteria growth in milk, he stated, show a relation to storage temperature, thus indicating the necessity for refrigeration. Milk is cooled at the farms and maintained below 40 F until it reaches the processing plant, where it is pasteurized by one of two methods: heating to 143.5 F for 30 min or to 161 F for 16 sec. Milk is refrigerated by chilled water cooled either by an ice builder-type or a service cooler refrigeration system.

who's doing what . . . Serving on the Nominating Committee are Harold McKee (Chairman), Hugh McMillan, Ray Ferguson, Otis G. Linimon and Ruben Zainfeld.

Discussed at the March 10 Board of Governors meeting were recommendations of the Nominating Committee for officers for the coming year. Suggested was advancement of the current officers, with the nomination of Morris Backer as Treasurer and Carl Daab, Jr. and William A. Jackson for the Board of Governors.

Speaking at the March 17 meeting, Region VIII Director W. J. Collins, Jr., discussed the proposed program for the Regional Meeting, held in Shreveport April 21 and 22. The Chicago meeting was reported on by Chapter Delegate Reg Taylor.

"Modern Trends in Commercial and Industrial Refrigeration" were detailed by guest speaker J. C. Lewis, with emphasis placed on development of the extended fin coil.

who's doing what . . . John Ames was nominated from the floor for membership on the Board of Governors for the 1961-62 season.

MICHIGAN . . . Solar transmission through glass, effective shading, heat absorbing glass, heat absorbing insulating glass and effect of these subjects on air conditioning loads were included in the talk of February speaker Otto F. Wenzler of Libby-Owens-Ford Glass Company, "Air Conditioning Loads Related to Glass Areas." Slide films were shown of several functional and experimental buildings using various kinds of shading devices, such as overhangs, screens, corridor-like exterior curtain walls constructed with two glass walls about two ft apart, and heat absorbing glass.

who's doing what . . . William Dull, Chairman of the local UEC Building Fund Drive, outlined plans for the campaign.

Speaker at the March 20 meeting was Harold E. Straub, Chief Research Engineer, Titus Manufacturing Corporation, who discussed "Principles of Room Air Distribution." Dealt with were factors affecting room air distribution during heating and cooling, with emphasis placed on four types of primary air outlet: floor, baseboard, sidewall and ceiling. The talk was based upon research conducted at the University of Illinois.

ILLINOIS-IOWA . . . Space travel was discussed at the February 23 meeting by John W. Massey of the George C. Marshall Space Flight Center. A major problem, he stated, is reliability of rocketry.

Proposed amendments to the Chapter By-Laws have been returned from the national By-Laws Committee with comments, which were read and discussed at the March 6 meeting of the Board of Governors. The proposed amendments will be revised and resubmitted.

Problems and suggested solutions in "Piping Expansion" were presented by R. M. Quick of Flexonic Corporation, guest speaker at the March 20 meeting. Types of expansion enumerated by him were: axial compression, lateral depression, angular rotation and combinations of these.

who's doing what . . . Nominated for Chapter officers are: D. J. Johnson, President; M. L. Smith, Vice President; P. J. Hannon, Secretary; John Sandberg, Treasurer; and J. Benbow and J. Louis, Board of Governors. Roy Conrad and Earl Beling have been appointed Chairman and Co-Chairman, respectively, of the local UEC Fund Raising Committee.

TOLEDO . . . Beginning his discussion by briefly explaining the function and operation of pneumatic transmission in control center work, March speaker R. J. Caffrey then spoke on the function of a control center. Extensive use of slides showed development of the control center from a standard temperature control panel to the counsel-type graphic control center.

Detailed in the April 3 talk of George F. Evans were ways in which the architect can cooperate with the engineer with regard to initial planning, space requirements, bid evaluation and coordination during construction.

who's doing what . . . Recommendations of the Nominating Committee are: President, J. S. Meyer; Vice President, R. C. Moorhead; Secretary, D. W. Dustun; Treasurer, G. W. Bleckner; and Board of Governors, N. W. Dawe, G. L. Heiser and O. Hockin.

EVANSVILLE . . . Uses of Cycolac plastics were shown at the March 7 meeting by speakers John M. Avery and John J. Haher. Physical and chemical properties of Cycolac and other plastics were compared.

Speaking on temperature control in the manufacture of missiles, Major General E. P. Mechling, U. S. A. F. (Ret.), touched upon the functions performed by the American Ordnance Association as related to our national defense. April speaker Mechling also discussed plans for offensive missiles, such as the Minuteman and Polaris.

who's doing what . . . Submitted by the Nominating Committee as officer-candidates for the 1961-62 season are: President, H. C. Shagaloff; 1st Vice President, C. L. Herndon; 2nd Vice President, D. S. Phillips; 3rd Vice President, D. Kuhlenschmidt; Treasurer, G. J. Ashcraft; Secretary, H. Tillman; Board of Governors, R. P. Garvey and I. J. Loeffler.

DAYTON . . . Preceding the February 23 meeting was a tour of the Moraine City Factory of Frigidaire Div, General Motors Corporation. Speaker of the evening was Richard R. Hough, Operating Vice President, Ohio Bell Telephone Company, who discussed "The Engineer's World of Tomorrow."

SOUTH FLORIDA . . . Cryogenics was the subject under discussion at a recent meeting, when J. R. McDonald and W. T. Rector of Air Products, Inc., were the guest speakers. Produced by this plant is liquid hydrogen for missiles. Materials used for low temperature work are stainless steel, copper and aluminum.

who's doing what . . . Comprising the Nominating Committee are Kelsey Sanders, J. Hendricks, Stephen Shelton, J. Middleton and Hugh Kirkpatrick.

Armand Cowan moderated a panel discussion on "Bid Shopping—Causes and Cures" at the March 14 meeting. Members of the panel were Everette M. Eignus of Edwin T. Reeder & Associates; Rod E. Overhold of M. R. Harrison Construction Corporation; Roy Barto of Atlas Air Conditioning Corporation; Henry Gaggstatter of R. L. Duffer Associates; and W. R. Ousley of Ousley & Associates, Inc.

who's doing what . . . Nominations for Chapter officers include: President, Armand Cowan; Vice President, James Beard and Hugh Kirkpatrick; Secretary, A. R. Dickterenko and Doug Howard; Treasurer, K. Cunningham and Oliver Parsons; Board of Governors, George Garrett, Aaron Hertz, John Lotz, Joseph Middleton and Robert Nichols.

EL PASO . . . Professor A. M. Lukens, former Head of the Mechanical Engineering Department at New Mexico State University, spoke at the February 20 meeting on "Solar Energy."

"Noise Control in Air Distribution Systems" was covered by March speaker R. Richardson of Tuttle and Bailey.

PUGET SOUND . . . Illustrating his talk with slides, March speaker H. B. Williams, Vice President in charge of engineering, McQuay, Inc., spoke on "Air-Cooled Condensers, Design, Selection and Installation."

who's doing what . . . Candidates for Chapter officers are: Keith Massart, President; Rod Kirkwood, First Vice President; David M. Hopkins, Second Vice President; Dean Moore, Secretary; Henry A. Bickel, Treasurer; and Dixon Ervin and Frank Nuyens, Board of Governors. Joseph Hubbard was nominated from the floor for Secretary.

FORT WORTH . . . "Human Factors and Human Engineering", a study of equipment design as it applies to the efficiency, safety and comfort of the individual, was discussed at the March 15 meeting by Dr. William G. Matheny of the Human Research Department of Bell Helicopter Corporation. Some aspects of environmental design of space capsules were covered, and good and poor procedures and practices were enumerated. Also examined was the relationship of human accidents and productivity to dry bulb and effective temperatures.

who's doing what . . . Jesse Mattox (Chairman), William Hulsey, Grant Johnson, Thomas B. Romine and Simeon T. Lake have been appointed to the Nominating Committee.

LONG ISLAND . . . .Speakers in a panel discussion on "Heat Transfer of Fin Type Coils" at the March 13 meeting were Alan Decker of Dunham-Bush, A. Magnani of Worthington and Eugene Ward of Carrier. Moderator was J. Mintin of Cool-Flo.

who's doing what . . . Hal A. Quick, B. Maxwell, Sy Williams, Ralph Caso, William Reichenback and A. Fuller have been proposed for the Board of Governors by the Nominating Committee.

Candidates for ASHRAE Membership

Following is a list of 138 candidates for membership or advancement in membership grade. Members are requested to assume their full share of responsibility in the acceptance of these candidates for membership by advising the Executive Secretary on or before May 31, 1961, of any whose eligibility for membership is questioned. Unless such objection is made these candidates will be voted by the Board of Directors.

Note: * Advancement † Reinstatement

REGION I

Connecticut

ABERCROMBIE, R. D., Partner, A & W Associates, Milford.
ALLEN, D. C.,* Vice-Pres. & Chief Engr., The Horton Co., Hartford.

Massachusetts

COOK, E. L., Editor, Boston Publishing Co., Boston.
DOUGHERTY, W. R. JR., Asst. Engr., Bethlehem Steel Co., Quincy.
KATZ, G. H., Self Employed, Medford.
MCDOUGALL, A. B., Sales Engr., Dunham-Bush Co., Quincy.
SEMPLE, J. W., Repr., American-Standard, Ind. Div., Newton.

New Jersey

BERNOFF, R. A., Dir. of Research, Melcor, Trenton.
GARGON, J. P., Estimator, Lee A. Dobson, Verona.
HAYES, F. X., Product Engr., Worthington Corp., A-C Div., East Orange.
MACINTYRE, N. J., Pres., Automatic Air Heating, Inc., Plainfield.
SAWKA, MICHAEL, JR., Supvsr. Bldg. Operations, The Mennen Co., Morristown.

New York

CAFFREY, R. J.,* Sr. Sales Engr., Johnson Service Co., Long Island City.
INCIARDI, J. P., Repr., Vibration Mountings, Inc., Corona.
KANE, R. G., Sales Engr., Robert B. Darling Co., New York.
KEENAN, G. J., Engr., Rochester Gas & Electric Co., Rochester.
MISHOE, G. H.,* Appl. Engr., C. W. Davis Co., Inc., Syracuse.
MUIRHEAD, BENJAMIN, Pres., B. J. Muirhead Co., Inc., Buffalo.
O'MAHONY, W. J.,† Engr., Peter Carver Assoc., New York.
PAULMANN, C. J., Sales Engr., Marlo Coil Co., New York.
RILEY, L. G., Engr., E. W. Tompkins Co., Inc., Albany.
STENNER, WILLIAM,* Mech. Engr., Bond Stores, Inc., Rochester, N. Y.
TOMPKINS, H. E. JR., Secy., E. W. Tompkins Co., Inc., Albany.

REGION II

Canada

ANDERSON, A. E., Gen. Mgr., Etna Commercial, Ltd., Winnipeg, Man.
BALDWIN, J. T., Repr., Tubecraft of Canada, Ltd., Toronto, Ont.
BULL, W. G., Associate, Yost Keen & Associates, Toronto, Ont.
BUTTS, H. W.,* Mgr. Mechanical Dept., Canadian Comstock Co., Ltd. Toronto, Ont.
CAMPBELL, G. C., Sales Engr., The Bird Archer Co., Calgary, Alta.
DINELLE, OSCAR, Estimating & Control, Ottawa Mechanical Services, Ltd., Ottawa, Ont.
ELLIOTT, JACK, Estimator, Goodram Bros., Hamilton, Ont.
GRIBBLE, K. R., Chief Draftsman, John H. Ross & Associates, Toronto, Ont.
HALDANE, G. P., Sales Repr., Fiberglas Canada Ltd., Toronto, Ont.
HANLEY, P. A., Jr. Engr., Breis, Frigon & Hanley, Ottawa, Ont.
HERBERT, K. E., Sales Engr., Michael Stuart Co. Ltd., Toronto, Ont.
HERSH, COLIN, Constr. Mgr., Beaver Engineering Co., Toronto, Ont.
KER, R. J.,* Divisional Mgr., Engineering Industries Co. Ltd., Montreal, Que.
LEICESTER, P. H.,* Engineer in Training, Angus, Butler & Assoc., Ltd., Edmonton, Alberta.
MA, W. Y., Design Engr., H. H. Angus & Associates, Toronto, Ont.
MURRAY, D. P., Chief Estimator, Watts & Henderson, Ltd., Toronto, Ont.
PAGEAU, RAYMOND, Vice-Pres., Mark Hot, Inc., Ville St.-Michel, Que.
REARDON, W. E., Mgr., Halliday Fuels, Ltd., Toronto, Ont.
STEIN, P. G., Dist. Mgr., Marley Canadian, Ltd., Montreal, Que.

REGION III

Maryland

BOLAND, L. J., Mgr., Trane Co., Bethesda.

Pennsylvania

CARRERAS, ROBERTO, Sales Engr., Frick Co., Waynesboro.
HOFFMAN, LEONARD, Project Engr., Merck Sharp & Dohme, Philadelphia.
NESHER, ALEXANDER,* Chief Engr., Barton Engineering Co., Philadelphia.
NICHOLAS, FRED,* Assoc. Professor, Engineering, Penn. State University, State College.
SONGSTER, W. F., Designer I, Catalytic Construction Co., Philadelphia.
SPIELVOGEL, L. G., Engr., Valley Engineering Co., Glenside.
STARK, J. F., Appl. Engr., Estimator, Frick Co., Waynesboro.

Virginia

GOLDBERG, R. F., Sales Engr., Powers Regulator Co., Arlington.
ROBINSON, C. J., Engr., Westinghouse Electric Corp., Staunton.
ROGAN, P. T., Br. Sales Mgr., Carrier Corporation, Norfolk.

REGION IV

Florida

LAYNE, J. W., Sales Repr., Trane Company, Miami.

North Carolina

FULBRIGHT, F. E., Repr., Atlas Supply Co., Winston-Salem.
NICKELL, G. T., First Vice-Pres., Air Conditioning Corp., Greensboro.

REGION V

Indiana

WITSKEN, C. H., Sales Engr., Welcraft Prod. Co., New Carlisle.

Ohio

HIGGINBOTHAM, D. C., Mgr. Wholesale Sales, Refrigeration Sales Corp., Cleveland.
SIPPEL, C. R., Project Engr., The Champion Paper and Fibre Co., Hamilton.
TURNER, A. E., A-C. Specialist, The East Ohio Gas Co., Cleveland.
WU, A. S., Engr., Gillmore Olson Co., Cleveland.
ZARNICK, B. F., Regional Sales Engr., Johns-Manville Sales Corp., Cleveland.

REGION VI

Illinois

BENDER, L. A., Cons. Engr., Skokie.

KUHN, O. F., Chief Mech. Engr., Sears Roebuck & Co., Chicago.
NISARGAND, U. L., Design Engr., U. S. Flex. Tubing Co., Bartlett.
THURMAN, A. L., Br. Mgr., American-Standard, Ind. Div., Moline.
ZOLADZ, E. J., Sales Engr., Dravo Corp., Chicago.

Iowa

CLARK, F. J., Gen. Mgr., Dunham-Bush, Inc., Marshalltown.

Michigan

GROSS, R. R., Supt. of Bldg., Detroit Public Library, Detroit.
POLKINGHORNE, R. E., Supervisor, Detroit Edison Co., Detroit.
SMITH, E. B., Estimator & Engr., Cloverland Contracting Co., Crystal Falls.
STOWELL, P. B., Pres., Chairman of Board, Leo A. Tilford, Inc., Jackson.

Minnesota

SARGENT, R. D., Pres. & Co-Mgr., Reader & Co., Inc., Pipestone.
STILKEY, D. C., Jr. Engr., McQuay Incorporated, Minneapolis.

Wisconsin

OVASKA, A. A.,* Asst. to Mgr. of Price & Data, Vilter Manufacturing Co., Milwaukee.

REGION VII

Alabama

FIVEASH, BROWNLEE, A-C. Sales, Alabama Gas Corp., Birmingham.

Louisiana

BOURGEOIS, M. J., Sales Engr., Equitable Equipment Co., New Orleans.
LAVIGNE, R. H., Sales Engr., R. K. Goode Co., Inc., Baton Rouge.
MELANCON, R. B., Repr., Harry Cash Company, Inc., Baton Rouge.

Missouri

LOCKWOOD, C. B., Vice-Pres., Treas. & Dir., Alco Valve Co., St. Louis.
MAY, D. H., Product Mgr., Granco Steel Products, St. Louis.
SCHWETYE, R. F., Asst. Prod. Mgr., Hussmann Refrigerating Co., St. Louis.
SEAT, W. W., Sales Engr., American-Standard, Ind. Div., St. Louis.

Tennessee

HOUSTON, S. D., Dist. Sales Mgr., Robert Shaw Cont. Co., Chattanooga.
QUISENBERRY, O. L., Design Draftsman, Tennessee Eastman Co., Kingsport.

REGION VIII

Arkansas

MCCORKLE, E. R., Field Engr., Johnson & Scott, North Little Rock.

Oklahoma

BOYD, J. Y., Sales Supvsr., Oklahoma Natural Gas Co., Oklahoma City.

Texas

DUBINSKI, G. Z., Secy.-Treas., Standard Electric Co., Inc., San Antonio.
FRAZE, M. L., Pres., Kieffer Plbg. & Htg. Co., Inc., Dallas.
HAYES, W. J., Vice-Pres., Kieffer Plbg. & Htg. Co., Inc., Dallas.
WOOD, H. B.,* Mfg. Repr., Bernhard Associates, Dallas.
ZERN, J. C., Sales Engr., Kotzebue Dist. Co., San Antonio.

REGION IX

Nebraska

CANON, J. J., Sales Engr., Carrier Air Conditioning Co., Omaha.
CONNOR, E. M. JR., Vice-Pres., Wray M. Scott Co., Omaha.
REDMOND, C. C., Owner, Best Plumbing & Heating, Beatrice.
SCHULTE, W. E., Appl. & Sales Engr., Omaha Public Power Dist., Omaha.
WOEST, R. F., Cons. Engr., Lincoln.

South Dakota

FODNESS, NORMAN, Plant Supt., Sioux Valley Hospital, Sioux Falls.

Utah

ROBINSON, C. K. JR., Sales Engr., Johnson Service Co., Salt Lake City.

REGION X

Arizona

ASH, J. L., Asst. Vtg. Engr., San Manuel Copper Corp., San Manuel.
CARRELL, G. P., Draftsman & Engr., John Paul Jones, Cons. Engr., Tucson.
ROGERS, G. M., Mech. Engr., Corps of Engineers, Tucson.
WILSON, R. S., Mgr., Hall Insulation Co., Tucson.

California

BANKO, A. H., Owner, Alex H. Banko, Inc., No. Hollywood.
BIEN, A. H., Engr., Isotherm Co., Bakersfield.
CASTRO, T. G., Design Engr., 6313th Cesron, Kadena Air Base APO 239, San Francisco.
DENHOLM, J. S. JR.,* Design Engr., U. S. Naval Shipyards, San Francisco.
DWYER, W. R.,† Br. Mgr., Norman S. Wright & Co., San Diego.
ESKRA, N. G., Refrig. & A-C. Sales, Ray Winther Co., San Francisco.
GARRISON, J. F.,* Design Engr., Daniel, Mann, Johnson, Mendenhall & Assoc., Los Angeles.
HILEMAN, C. W., Contract Sales Mgr., Powers Regulator Co., Los Angeles.
MANES, P. R.,* A-C. Engr., Norris-Thermador Corp., Los Angeles.
MCLAUGHLIN, C. E.,* Sales Engr., American-Standard, Ind. Div., San Francisco.
MELSHEIMER, TED, Owner, T. Melsheimer Co., Torrance.
MORRIS, J. I., Project Engr., Consulting Engineers Inc., San Diego.
MURPHY, J. E., Sales Engr., Powers Regulator Co., Sacramento.
SHERIFFS, LEIGH,* Associate Chief Engr., Lawrence B. Freed, Beverly Hills.
SYLVANUS, A. B., Architect, Engr., Ets Hokin & Galvan, San Diego.
WOERNER, P. W., Sr. Appl. Engr., Worthington Corp., Alhambra.

Washington

ANTLES, L. S.,* Engr., John Graham & Co., Seattle.

FOREIGN

Australia

GORMAN, R. K., Partner, Kevin Gorman & Partners, Melbourne, Victoria.

England

THOMPSON, E. W., Engr., Weatherfoil Ltd., Slough, Bucks.

France

SEBEO, JACQUES, Appl. Engr., Anemotherm, Vanes (Seine).

Germany

ZNAKOVSZKY, ZOLTAN, Mfg. Repr., Berlin-Grunewald.

Hong Kong

WU, M. C., Engr. in Charge, Winsome Company, Hong Kong.

India

PASSEY, A. D., Teacher Trainee, Indian Institute of Technology, Kharagpur.

Iran

GHARABEGIAN, S. K., Design & Site Engr., Shelleh Kavar Co., Tehran.

Italy

COSTANTINO, MARIO, Export Mgr., Aster S.p.A., Milan.
MAZZINI, ANGELO, Mfg. Repr., Milan.

Jamaica

NOVAK, J. C., Cons. Engr., Ewbank & Ptns., Kingston.

Lebanon

GUBER, NIKITA, Field Engr., Hupp International, Beirut.

Pakistan

BELGAUMWALA, J. A., Installation Supvsr., Jaleel Brother Ltd., Karachi.

Philippines

REYES, A. V., Chief Engr., Pacific Star, Inc., Quiapo, Manila.

Singapore

YIN, C. K., Instructor, Singapore Polytechnic, Singapore.

Turkey

URAN, N. F., Engr., Nejdet Uran Engineering Co., Ankara.

United Arab Republic

SHOUKRY, MOSSAAD, Construction Supt., Koldair, S.A.E. Cairo.

West Indies

CARRINGTON, A. G., A-C. & Refr. Engr., The Emtage Electrical Co., Ltd., St. Michael, Barbados.

STUDENT

ROLL, W. P. JR., University of Houston, Houston, Texas.

ASHRAE STANDARDS PROJECTS

STANDARD No.	PROJECT COMMITTEES FOR PROPOSED REVISIONS			
16-56R	**Method of Testing for Rating Room Air Conditioners**			
	Chairman: R. H. Meyerhans *Liaison:* J. R. Schreiner	*Members:* C. G. Coyne K. D. Crook R. W. Gilmer	E. Gmoser P. E. Kolb J. R. Schreiner	E. H. Schwenker E. C. Tanner
17-48R	**Methods of Testing for Rating Thermostatic and Constant Pressure Refrigerant Expansion Valves (ASA B60.1-1950)**			
	Chairman: D. C. Albright *Liaison:* T. B. Simon	*Members:* F. Y. Carter K. E. Wilson W. F. Wischmeyer	*Alternates:* W. Stafford D. S. Sterner	
18-56R	**Methods of Testing for Rating Self-Contained Mechanically-Refrigerated Drinking-Water Coolers**			
	Chairman: W. Taylor *Liaison:* T. B. Simon	*Members:* F. O. Graham P. R. Lynn	F. J. Reed	E. W. Scott
33-58R	**Methods of Testing for Rating Forced-Circulation Air-Cooling and Air-Heating Coils**			
	Chairman: S. W. Anderson *Liaison:* K. O. Schlentner	*Members:* R. D. Blum W. C. Dackie G. J. Haufler	J. J. Manning F. W. McKenna O. J. Nussbaum	D. D. Wile H. B. Williams
53-34R	**Method of Testing for Rating Steam Unit Ventilators**			
	Chairman: J. W. McElgin *Liaison:* K. O. Schlentner	*Members:* L. A. Cherry A. F. Hubbard	D. H. Krans	L. G. Miller

PROJECT COMMITTEES FOR PROPOSED STANDARDS

31P	**Method of Testing for Rating Solenoid Valves for Refrigeration and Air Conditioning Systems**			
	Chairman: J. A. Schenk *Liaison:* J. Klassen	*Members:* D. C. Albright F. Y. Carter C. C. Hansen, III	D. S. Sterner W. F. Wischmeyer	*Alternates:* J. M. Strauss K. E. Wilson
36P	**Methods of Equipment Sound Testing**			
	Chairman: C. M. Ashley *Liaison:* S. P. Soling *Consultant:* R. J. Wells *Secretary:* W. R. Kerka	*Members:* W. E. Blazier J. B. Chaddock J. B. Graham	A. F. Hubbard D. H. Krans	S. P. Soling
39P	**Methods of Testing for Rating Unitary Heat Pumps for Air Conditioning**			
	Chairman: G. L. Biehn *Liaison:* J. R. Schreiner *Secretary:* R. N. Mahendra	*Members:* C. H. Belt H. W. Jobes	R. C. Lower R. G. McCready	W. A. Spofford
41P	**Standard Measurements**			
	Chairman: C. W. Phillips *Liaison:* F. J. Reed	*Members:* F. J. Reed	W. A. Spofford	G. Thompson
51P	**Testing Air Moving Devices**			
	Chairman: S. Konzo *Liaison:* H. T. Gilkey	*Members:* L. G. Seigel	G. L. Tuve	D. D. Wile
52P	**Standard for Evaluation of Air Cleaning Devices for General Ventilation and Air Conditioning**			
	Chairman: P. R. Achenbach *Liaison:* H. T. Gilkey	*Members:* W. C. L. Hemeon	R. F. Logsdon	P. E. McNall
54P	**Method of Testing for Rating Gravity Roof Ventilators**			
	Chairman: R. G. Nevins *Liaison:* J. Klassen	*Members:* G. C. Breidert F. N. Calhoun R. B. Engdahl	W. V. Hukill J. P. Johnson K. E. Robinson	F. B. Rowley L. J. Shumaker
55P	**Thermal Comfort Conditions**			
	Chairman: R. G. Nevins *Liaison:* F. J. Reed			

ROOM AIR CONDITIONER STANDARD

Available for Review

A. T. BOGGS, III
ASHRAE Technical Secretary

According to the By-laws, review copies of prospective standards are made available for industry comment for a period of 60 days. The proposed revision to that part of Standard 16 concerning room air conditioners has been approved by the project committee and the Standards Committee. Review copies are available from the technical secretary for 60 days. Comments should be submitted to the technical secretary before July 7.

The following standards have recently been approved by the Board of Directors, having previously been approved by the Standards Committee and made available for industry review for a period of 60 days: 22-61 – Methods of Testing for Rating Water-Cooled Refrigerant Condensers: 24-61 – Methods of Testing for Rating Liquid Coolers; 40-61 – Methods of Testing Heat Operated Unitary Air Conditioning Equipment. Copies of Standards 22 and 24 are available from the Sales Department at 75c per copy. Copies of Standard 40 are available at $1.50 each. Effective date of these Standards is April 3.

One of the basic standards for the air-conditioning industry was ASRE Standard 16. This standard covered all types of air-conditioning equipment and the Standards Committee believed separate standards on room air conditioners and unitary equipment would be more helpful to industry. For this reason, the revision to Standard 16 included separation into the following standards: 16 – Room Air Conditioners; 37 – Unitary Equipment; 39—Heat Pumps; 40—Unitary Heat-Operated Equipment. Standard 37 on unitary equipment and Standard 40 on unitary heat-operated equipment are available at the present time. Standard 39 on heat pumps is being reviewed by industry and is ready for consideration by the Board of Directors. With the announcement in this issue of review copies of Standard 16, the Standards Committee expects to have all four parts of the original standard available for industry before September.

ASA: The 12th National Conference on Standards will be held October 10-12 at the Rice Hotel in Houston, Texas.

Sectional Committee Y14 – Draft copy of Power Switchgear and Industrial Control Section, of American Standard Y14.15-1960, Electrical Diagrams, is being distributed to industry for comment. Copies are available for comment from Standards Manager, ASME, 29 West 39th Street, New York 18.

ASTM: Standards on Cement—The 1960 edition of the **Compilation of ASTM Standards on Cement** is now available. This reference contains 8 specifications, 26 methods of test, and several definitions. Two specifications for laboratory apparatus are also included. Copies of this book may be obtained from ASTM headquarters, 1916 Race Street, Philadelphia 3, Pa., at $4 each.

The 1960 edition of the **Compilation of ASTM Standards on Gypsum Products and Plaster Aggregates** is now available. The volume contains 16 specifications, 11 methods of test, 4 sets of definitions related to gypsum products and plaster aggregates as well as related standards. Copies of this book may be obtained from ASTM at $2.75 each.

GSA: The General Services Administration has issued a new Federal interim specification for **Flexible Unicellular Thermal Insulation** dated February 24, 1961. The material is described in the specification as a flexible unicellular composition such as an expanded synthetic elastimor, suitable for use as thermal insulation and condensation control. The maximum upper temperature use limit shall be 200 F, and is intended for use on both lines and equipment. The specification therefore covers the material in both tube and sheet form. The Flexible Unicellular Manufacturers Task Force is located at 342 Madison Avenue, New York 17, N. Y., with A. L. Faubel as acting secretary. Copies of the new GSA specification are available from the Task Force Office or from GSA offices.

MCA: The Manufacturing Chemists' Association announces the availability of revised editions of four **chemical safety data sheets.** These booklets cover benzene, nitric acid, paraformaldehyde, and anhydrous ammonia. In addition to the characteristics of these chemicals, the following procedures are included: Hazards, engineering control of hazards, employee safety procedures, fire fighting techniques, recommended handling and storage practices, tank and equipment cleaning and repair procedures, waste disposal, medical measures necessary and first aid requirements. For further information contact MCA at 1825 Connecticut Avenue, NW, Washington 9, D. C.

UEC Will House World's Outstanding Engineering L

J. H. CANSDALE
Assistant Secretary
Public Relations
and Fund Raising

A few short years ago, the United Engineering Center was the dream of a few visionaries. Today, as can be seen from the illustration, it is rapidly nearing completion. Occupancy by the staffs of 19 engineering societies, including ASHRAE, is scheduled for August of this year.

One of the facilities of the Center, which can be shared by every ASHRAE member no matter where he resides, will be the Engineering Societies Library (ESL). ESL is nearly fifty years old, but its roots go back much further; for it was originally formed by combining the separate libraries of the ASCE, AIME, ASME and AIEE.

PUBLICATIONS FROM ASHRAE

Those societies moving into the UEC may add their engineering books and technical papers to the ESL. The Library wants ASHRAE technical material, too. As the ESL will become your library, it is important that it have complete files of ASHRAE publications.

ASHRAE currently maintains a library at its headquarters at 234 Fifth Avenue. Included are bound sets of the JOURNAL, TRANSACTIONS, GUIDE AND DATA BOOKS and periodicals from the present society and predecessor societies. Also stocked is a wide variety of textbooks and publications related to our industry.

ASHRAE plans to discontinue its own library as such and to retain only a limited number of publications. This action is wise and significant because all societies occupying space in the new Center will pro-rata contribute to the support of the ESL.

COLLECTION AND STAFF OF ESL

A Library, like a waterworks, can meet all needs only when its intake is adequate and continuous, when its reservoir is ample, and its service facilities are well equipped and staffed. The ESL has a selected collection of 180,000 volumes, 27,000 maps, 10,000 indexed bibliographies, all major and many minor indexing and abstracting services in engineering, physical sciences, and technology, and many thousand unpublished papers of its supporting societies. Incidentally, this means that it has some six miles of books to move into the UEC. The Library is currently receiving 3400 periodicals and other serial publications from 45 countries, in 22 languages.

In addition to collecting extensively, but selectively, engineering publications of all types on a world-wide basis, the ESL maintains extraordinarily complete files of all technical publications and papers of its supporting societies. Its collection of unpublished papers is particularly important, for many of them are manuscript copies not available elsewhere, not even in the societies' own files. The Library has special card indexes to some sets of these publications. It also, in some instances, cooperates with the editorial staffs of its supporting societies in the preparation of published indexes.

After preprint and published stocks of their publications are exhausted, the societies refer inquirers to the ESL for photoprint or microfilm copies. Often this is done simply by transferring the inquirer's letter or order directly to the Library. This seems to be the best way to serve with the least possible delay.

During the past year, 9 per cent more volumes were added than in the previous year. The accessions rate is 40 per cent greater than three years ago. The Library has a staff of 30 persons; half are clerical and half are trained librarians with technical training and experience, and knowledge of at least a dozen foreign languages.

SERVICES AVAILABLE

The services of the ESL include a reading room open six days each week all year, and five nights a week except during the summer.

Thousands of requests from members for brief information which can be located readily are answered without a charge. For ASHRAE members and others requiring extensive information, literature searches and bibliographies will be made, for a fee, to the specific requirements of the inquirer. The service ranges from recommending some books on a specific subject to the preparation of comprehensive annotated bibliographies of books, articles and reports. Searches are also made for disclosures related to patents. All search work is kept confidential.

The Library Staff also prepares bibliographies on subjects of general engineering interest. These may be purchased by anyone. A list is available on request.

Translations of engineering and technical articles are made from all languages into English by "consultant" translators who are familiar with engineering. All translations are reviewed by a member of the Staff of the Library to assure accuracy of translation and the quality of the English. Photoprint and microfilm copies of material in the Library are made on request.

In each recent year about 700 engineering books have been reviewed by the Library Staff. The reviews are made available to the editors of the journals of the societies which support the Library. The editors select and publish those they consider to be of interest to their members and other readers so they may learn about recent books in their field. Most of the books reviewed, as well as others in the Library, may be borrowed by members of supporting societies.

All of these services, except loans of books to members, are available to anyone. They are used by engineers, scientists, technologists and industry in this country and throughout the world.

In each recent year many hundreds of letters have been referred by the headquarters staffs of supporting engineering societies to the ESL for reply, thus saving the societies time and money.

Library

ESL AND THE ENGINEERING INDEX

Through cooperation between these two separate organizations, engineers and industry have services unmatched elsewhere in the world, i.e. an extensive index published on cards and also in an annual volume which covers the contents of the largest engineering library in this country, intended primarily to serve graduate and practicing engineers.

All publications received and retained by the ESL, including the 3,400 currently received periodical and other serial publications, as well as a substantial number of bulletins, reports, symposia and unpublished papers are made available to the Staff of the Engineering Index, Inc. (EI). From these publications, 33,000 articles and other items were indexed in 1960. Each reference is annotated and all are published in a daily card service and subsequently in a cumulated annual bound volume. The card service is also available on a weekly basis in any one or more of the 255 subject divisions. Inquiries and subscriptions for the EI services should be addressed to the Engineering Index, Inc., which now is at 29 West 39th Street, New York 18, N. Y., but which will move into the United Engineering Center this summer.

Inasmuch as all indexed material is retained by the ESL, there need be no question about where to find the original of any articles indexed in the EI. The articles may be read at the ESL, which is open to anyone, or a photoprint or microfilm copy may be ordered.

GROWTH OF USE OF ESL

The ESL is a busy information center which becomes busier each year. Last year the use of the Library increased 14 per cent over the previous year. Use of the reading room has remained relatively constant since its peak during the depression years of the 1930's. On the other hand, non-visitor use of the Library by mail, telephone and telegraph increased 26 per cent in the last year; 176 per cent in the last ten years. The total number of those served by the ESL was nearly 60,000, of which 68 per cent did not visit the Library. A sampling of 15 per cent of last year's correspondence showed users in 31 foreign countries, and in 47 of the 50 states of the U.S.A.

The ESL is used by individuals and organizations. Of the latter, those with libraries of their own are among the greatest users. Last year on 6334 orders, 87,500 photoprints were supplied – an increase of 23 per cent; 484 microfilm orders were filled – up 25 per cent; 168 literature searches were made – up 29 per cent; 715,000 words were translated from a dozen foreign languages – up 84 per cent; 2270 books were lent to members – up 13 per cent; and 24,700 telephone inquiries were answered – up 36 per cent. Income from the paid services of the ESL increased over 40 per cent. Last year the Founder Societies' direct contribution to the Library amounted to about 58 per cent of its total income. Most of the other 42 per cent came from charges for services, and smaller amounts from endowments, royalties and contributions from other societies.

LOOKING TOWARD THE FUTURE

There was a time when an engineer might have in his head and handbooks most of the information he might need. Today that is seldom possible. The use of rule of thumb methods is disappearing. Frequently there must be research. Needed information can be found in many instances through literature research, which generally is less costly than original research. No library can claim to be able to meet all needs, but many of the engineer's needs can be satisfied through use of the ESL.

Rapid growth in use of the ESL indicates a growing awareness of its value to the engineering profession. With the larger number of engineering societies joining together in the United Engineering Center in active support and use of the ESL, it is anticipated that it will be of increasing service to the engineering societies and profession. A free information folder about the Library and its services, and a list of available bibliographies are available on request to the Engineering Societies Library, 29 West 39th St., New York 18, N. Y.

REAFFIRM YOUR FAITH IN THE ENGINEERING PROFESSION BY CONTRIBUTING TO THE UNITED ENGINEERING CENTER BUILDING FUND. SEND IN YOUR CONTRIBUTION, TODAY.

BULLETINS and CATALOGS

Collector. Described in this bulletin are improved design cyclone collectors, with emphasis placed on changes in collector tube and vane design. Dimensional data, efficiency curves and volume-draft loss nomographs are included.
John Wood Company, Air Pollution Control Div, Nicolet Ave., Florham Park, N. J.

Boiler. Bulletin 1275 describes and illustrates the new Powermaster positive flow boiler. Drawings and photographs show such innovations as: positive circulation design, with side located furnace; hinged front and rear covers and hinged dry back and optional wet back construction. Sizes, ratings and dimensions are included, as well as approximate fuel consumption for each size.
Orr & Sembower, Inc., Reading, Pa.

Silver Brazing Alloys. Fully illustrated and diagrammed, 24-page Bulletin ADC847C discusses brazing procedures, problems and solutions. All silver brazing alloys, from those in standard rod forms to Fluxcor 45, which incorporates the fluxing agent in the center of the wire, are described extensively, stressing advantages and applications.
Air Reduction Sales Company Div, Air Reduction Company, Inc., 150 E. 42nd St., New York 17, N. Y.

Room Air Conditioners. Specifications for 20 new 1961 models are listed in four-page Bulletin LL-471, which also pictures each of the four series and includes a room air conditioner selection guide. Illustrated also are a new window mount and its in-wall sleeve.
Chrysler Corporation, Airtemp Div, P. O. Box 1037, Dayton 1, Ohio.

Combustion Controls. Described in 20-page Bulletin 1002 is an expanded product line of electronic combustion controls, including oxygen analyzers, flow meters, draft gauges, control valving, interlock switches and flame sageguard equipment. Systems engineering facilities at the company are discussed, as well as new developments.
Reliance Instrument Div, Electro-Mech Corporation, Norwood, N. J.

Residential Heating. In loose-leaf binder form, this catalog is divided into twelve sections, each of which contains product information, technical data, installation instructions and piping and wiring diagrams on one of the company's lines of residential heating equipment. Covered are hydronic and warm air units for both oil and gas-firing, hydronic and electric baseboard and a DeAirator-Expansion Tank, among others.
General Automatic Products Corporation, 2300 Sinclair Lane, Baltimore 13, Md.

Glass. Recently published, a revised edition of "This Is Glass" provides a comprehensive discussion of glass and glass-ceramics. Illustrated, the 68-page booklet reviews the history of glass and details basic types. Described also is the expanding role of glass in science, industry, electronics, lighting and the home. Included is a section on new glass-ceramic materials and a two-page chart giving properties of selected glasses and glass-ceramics.
Corning Glass Works, Corning, N. Y.

Compact Valves. Provided in the Compact Capacity line are streamlined valves to meet most refrigeration and air conditioning requirements. Small in size, they are designed to be used for original equipment installations or for replacement. Descriptive of the line is six-page Bulletin 262-B.
American Radiator & Standard Sanitary Corporation, Controls Div, 5900 Trumbull Ave., Detroit 8, Mich.

Dry-Type Air Filter. Subject of eight-page Bulletin 228 is the Dri-Pak dry-type unit air filter, which inflates when the ventilation system is in operation and deflates when the system is shut down. Described are uses, construction, installation, operation and holding frame arrangements. Also included are dimensional drawings, performance data, space requirements and suggested specifications.
American Air Filter Company, Inc., 215 Central Ave., Louisville 8, Ky.

Control Valves. Available in cast steel as Model 670 and in stainless steel as Model 770, valves covered in Flyer JNP-7 are manufactured in ¼, ⅜, ½, ¾, 1 and 1¼-in. capacities. Suitable for an extensive range of steam, air, water, oil and gas applications, units are self or controller-operated. Featured in the bulletin are materials of construction, sizing charts, cutaway view, flow curve, dimensional drawing, explanation of sliding gate seat, ranges, ratings and extensive engineering specifications.
OPW-Jordan Corporation, 6013 Wiehe Rd., Cincinnati 13, Ohio.

Side-Fired Boilers. Describing ST series boilers, in which the burner is mounted at the side to reduce floor space requirements, four-page Bulletin A-103 lists specifications for 6, 12, 15, 20 and 30-hp, oil, gas or combination gas and oil-fired boilers, with 201,000 to 1,005,000-Btu/hr capacity at 100 to 125-psi working pressure standard. Also discussed are special heat extractors installed in firetubes to control the rate of hot gas flow.
Eclipse Fuel Engineering Company, Boiler Div, Manufacturers Rd. & Compress St., Chattanooga 5, Tenn.

Pipe and Tubing Measurement. Standard methods of measurement are supplied by 16-page Bulletin 631, which discusses both similarities and differences in measuring various fittings and fluid carrying lines. Also covered are standard practices of measuring adapters and complete assemblies. Dash numbering systems commonly used in pipe and tubing measurements are explained in chart form.
Aeroquip Corporation, 300 South East Ave., Jackson, Mich.

Circulating Water Treatment. Containing no chromates or acids, Air-Con Emulsion is a concentrated liquid treatment that works through colloidal action rather than the softening of water to remove and prevent rust and scale formation in air conditioning and open circulating systems. Descriptive of the process is a four-page bulletin.
American Sand-Banum Company, Inc., One N. Merrick Ave., Merrick, New York.

Contactors, Starters. Eight-page Bulletin GEA-7316 describes an extensive line of contactors and starters for air conditioning and refrigeration. New 30 and 40-amp units that have been added recently to this line of compact controls are pictured and discussed, together with 50 and 60-amp models. Also described are overload relays, custom control panels and a step starting accessory available for air conditioning applications.

Outline drawings and dimensions are given for both open and enclosed contactors and starters, together with

a page of information that includes load and coil data, outline numbers, wiring diagrams and weights of units. A selection guide lists nomenclature and current ratings for the line.
General Electric Company, One River Rd., Schenectady 5, N. Y.

Engineering Manual. Available from this company is an engineering manual on radiant heating, cooling and acoustic ceilings. Contained are more than 200 pages of data, drawings and specifications, with additional material to be released at regular intervals. Price is $40.00.
Burgess-Manning Company, Architectural Products Div, Libertyville, Ill.

Selection Guide. Simplified instructions for determining room-by-room residential heat loss, sizing the boiler and selecting radiation are given in this ten-page Hydronic Fitters' Guide. Included are a Table of Residential Heat Loss Factors for walls, ceilings, floors and infiltration and a simplified Table of Heat Losses in Btu/hr, based on a temperature difference of 70 F. A Table of Conversion Factors makes it possible to determine the heat loss for any temperature difference from 20 to 110 F.
Crane Company, P. O. Box 780, Johnstown, Pa.

Humidity Control. Included on the contents page of 72-page Catalog 661 are such headings as: Humidity measurement and control systems; electric hygrometer instrumentation—sensing elements, indicators, recorders, controller, dew point measuring instruments, cable selection charts and procedure for developing hygrometer systems.

Extensively illustrated, the bulletin contains charts and descriptive information on many different types of humidity control equipment, as well as a temperature conversion table.
Hygrodynamics, Inc., 949 Selim Rd., Silver Spring, Md.

Packaged Burners. For larger commercial and industrial applications, Series R burner packages, described in Flyer B18/5, are offered in a range of inputs from 4,000,000 to 70,000,000 Btu/hr. Units are illustrated and arrangement, draft, combustion controls, operating cycle, electronic combustion safeguard and start-up service are discussed.
Webster Engineering Div, Midland-Ross Corporation, Box 15709, Tulsa, Oklahoma.

Mixing Heads. Polyurethane foam producing equipment is the subject of Flyer L-236-1, which lists specifi-

cations and information. Units are offered for two or three-component systems and systems requiring up to six separate components.
Martin Sweets Company, Inc., 114 S. First St., Louisville 2, Ky.

Air Purification. Using activated charcoal, air purification equipment listed in four-page Bulletin T-264 includes adsorber cells, circulating units and disposable filters. Circulation rates, dimensions, resistance, nominal air flow and capacity in cfm are among data given.
Barnebey-Cheney Company, Cassady at 8th Ave., Columbus 19, Ohio.

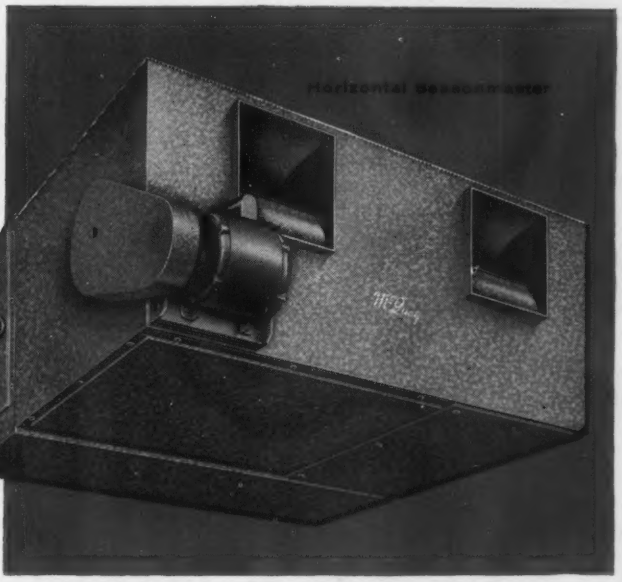
Control-Indicator. Descriptive of the Vis-a-Trol automatic pump control and liquid level indicator is four-page Bulletin AC-100. For submersible and vertical-type sewage and sump pumps and septic tank effluent pumps, the unit permits locating the control panel for the pumps at the sump location or at any remote point up to 300 ft distant.
Weil Pump Company, 1530 N. Fremont St., Chicago 22, Ill.

Tank Heaters. Steel, cement-lined, copper-lined and solid copper-silicon hot water storage tanks, together with an extensive range of heating elements for these tanks, are the subject of 16-page Catalog 602. Piping diagrams, engineering data and heating capacity charts are presented.
Old Dominion Iron & Steel Corporation, Belle Isle, Richmond 3, Va.

Axial-Flow Fans. Units for heating, cooling, ventilating, fume removal and drying systems are described in 16-page Bulletin AFF-61. Covered are fans with direct-connected motors and designs with V-belt drive. Features, construction details, optional equipment, specifications and tables of capacities and dimensions are included.
L. J. Wing Manufacturing Company, 2300 N. Stiles St., Linden, N. J.

Control Valves. Designed for application to year-round air conditioning systems using hot or cold water as a heat transfer medium with a single supply line and one return line, Selectaflow self-contained, thermostatic selector valves are the subject of eight-page Bulletin 272-A. Specifications on both direct and remote actuator types are listed. Heating and cooling cycle operations are illustrated by cross-sectional schematic diagrams. Also presented are brief data covering Winterflow-Summerflow

(*Continued on page* 100)

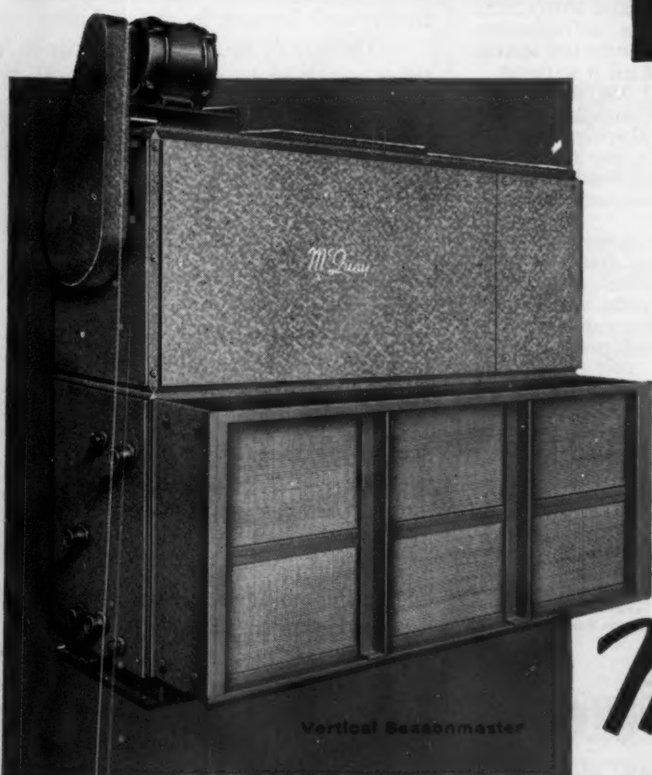

Horizontal Seasonmaster

McQuay SEASONMASTERS

NEW single and multi-zone central station air conditioning units

SAVE valuable SPACE


Multi-Zone Seasonmaster


Vertical Seasonmaster

Compactness saves space, and McQuay proudly offers a new line of quality engineered central station air conditioning units that do just that. A new space saving design, available in both single zone draw through and multi zone, or double duct, blow through arrangements in low pressure (up to 3" s.p.) and medium pressure (3" to 5½" s.p.) makes the McQuay line the most complete and versatile in the industry. Thirteen single zone and eleven multi zone models are available. See the McQuay representative in or near your city, or write for catalog 560 to McQuay, Inc., 1606 Broadway N. E., Minneapolis 13, Minnesota.

McQuay INC.

AIR CONDITIONING • HEATING • REFRIGERATION

MAKE THE SEASONS COME TO YOU

BULLETINS

(*Continued from page 96*)

single-purpose controls, a selection chart and other components and accessories for the air conditioning industry.
American Radiator & Standard Sanitary Corporation, Controls Div, 5900 Trumbull Ave., Detroit 8, Mich.

Fuel Oil Treatment. Designed to minimize slag and corrosion in boiler systems, this residual fuel oil treatment is described in Flyer El-159. How the product improves heat transfer and protects the convection and cold end of boiler furnaces is outlined. A case study and methods of application also are included.
Nalco Chemical Company, 6216 W. 66th Pl., Chicago 38, Ill.

Air Conditioning Lines. Covered in four-page Bulletin GAC 150.01 are low cfm and heavy-duty fan-coil units, remote and multizone-type central station air conditioners, sprayed coil dehumidifiers, high pressure central station units, heating-ventilating units, evaporative and air-cooled condensers and extended-surface heating and cooling coils. Units are shown and concise application and specification data are given.
Drayer-Hanson Div, Hi-Press Air Conditioning of America, Inc., 3301 Medford St., Los Angeles 63, Calif.

Gas-Fired Burners. Presented in 14-page Bulletin H-1 are eight atmospheric, gas-fired burners with maximum firing capacities from 10,000 to 2,750,000 Btu/hr. Provided are dimensions, specifications, engineering data and ordering information for pipe, wheel, immersion, ring, box and U-type burners, atmospheric injectors and packaged burners. Burner accessories include gap-type burner nozzles, mounting cages and a draft-compensating pilot.
Eclipse Fuel Engineering Company, Rockford, Ill.

MTW SYSTEMS

(*Continued from page 52*)

the valve at design flow which is a significant percentage of the maximum available head, are also necessary. Outdoor reset of water temperature improves control, and should be used wherever possible. Secondary pumping, by providing hydraulic isolation of the terminal, also improves the quality of control. These factors are comparatively well known in the industry.

The use of smaller water quantities can lead to more economical systems and better control. But the smaller the design water quantity with respect to the load, the more precise the engineer must be in his system design. With conventional design, the capacity flow curve is comparatively flat at design conditions, and little difference in total capacity results if the system actually operates anywhere from 80 or 120% of the calculated flow (controllability at reduced loads is another matter). With smaller water quantities, however, larger changes in maximum capacity will accompany flow errors of the same percentage. Fig. 3 illustrates this point. A design for small water quantities can return rich rewards in economy and better performance. But, it cannot be achieved by rule of the thumb. It requires that the designer truly engineer the system.

Today's Topic: Refrigerator Shelves

How do yours rate with the Board of Public Opinion?

Rusty refrigerator shelves get a thumbs-down verdict; so do shelves and trays that warp and crack. But this is the risk when you replace quality *aluminum shelves* with those made of other materials. The few pennies you save per refrigerator can be the costliest savings ever when you evaluate the possible long term damage to your reputation . . . and sales!

Aluminum shelves and trays are a basic requirement in *every price range*. Only aluminum is rustfree and carefree, lightweight and strong, nontoxic and corrosion resistant. So why take a chance? Why gamble with your reputation for quality when you can be so sure with aluminum.

Aluminum shelves can be easily fabricated in many cost-reducing forms—from wire, formed sheet, expanded extrusions, etc.—in a variety of alloys and colors. Reynolds Aluminum Specialists will help in your design and production—or will fabricate shelves for you to your specifications. *Reynolds Metals Company, P. O. Box 2346-AP, Richmond, Virginia.*

REYNOLDS ALUMINUM

Watch Reynolds TV show "Harrigan & Son", Fridays—ABC-TV

PARTS AND PRODUCTS

PRECIPITATOR

Custom built to suit the job, these electronic precipitators use modular construction techniques. High efficiency is achieved through use of a

flat collector plate design. Pockets in the plate trap and hold the dust, reducing the chance of re-entrainment. Non-welded, roll-formed plate construction minimizes the possibility of warping and produces optimum flatness, which contributes to higher efficiency. Four types of high voltage rectifier are available: silicon, selenium, high voltage vacuum tube or mechanical. Control can be either a saturable reactor, monocyclic network or a resistor type.

American Radiator & Standard Sanitary Corporation, Industrial Div, Detroit 32, Mich.

REPLACEMENT MOTOR

Mounting dimensions can be varied up to one in. with an adjustable mounting adapter on this new permanent-split capacitor and shaded-pole replacement motor line. Varying the mounting dimensions means that the 24 models in the line can replace most of the 5½-in. diam fan motors used in room air conditioners and many on furnace blowers.

Location of the adapter is on the motor's extended through-bolts, so that it can be adjusted axially along the motor shaft. The adapter can be omitted for band-mounting of the

(*Continued on page* 105)

LONG RANGE PLANNING

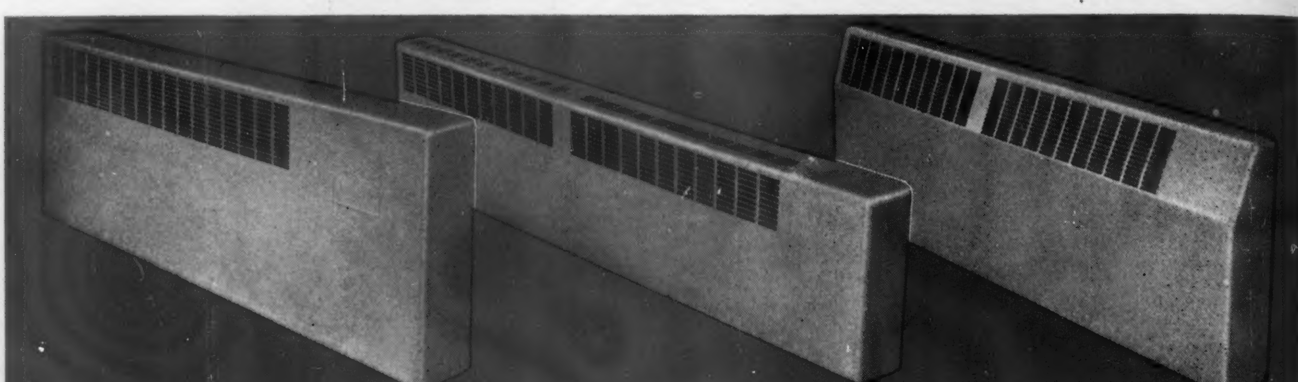
(*Continued from page* 57)

experience in the Society. Certainly, this points to past presidents, who know the operation of the Society most intimately. It is understandable then why 12 of the 17 members have been past presidents or became presidents. All of them had experience on the Board of Directors prior to becoming members of the Committee.

The present members of the Committee are: Past Treasurer J. Donald Kroeker, chairman; Past President Dan D. Wile, vice chairman; Frank H. Faust; Past President P. B. Gordon; Past President Elmer R. Queer; and Si J. Williams, Jr.

At the annual meeting of 1960 in Vancouver, B. C., the Board of Directors assigned comprehensive studies on research, on publications, organization and finances of the Society. The report on research was adopted by the Board of Directors, as discussed above. Studies are under way on publications and organization. Reports on immediate problems in these areas will be made to the Board of Directors at the Denver 68th Annual Meeting in June, with comprehensive reports expected for the Semi-annual Meeting in St. Louis next January. The study and report of finances, dependent largely on the studies on publications and organization, as well as research, is expected to follow for the 69th Annual Meeting.

"Slide 'n Snap" FIN VECTOR RADIATES DESIGN VERSATILITY



NOW... DUNHAM-BUSH OFFERS...

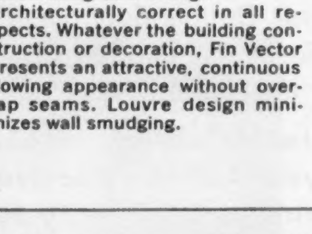
CHOICE OF...FRONT COVER...FRONT/TOP COVER...SLOPING COVER

Completely refashioned to architects' and contractors' specifications, Dunham-Bush Fin Vector full "wrap-around" cover in all three styles introduces exciting possibilities for heating industrial, commercial and institutional buildings.

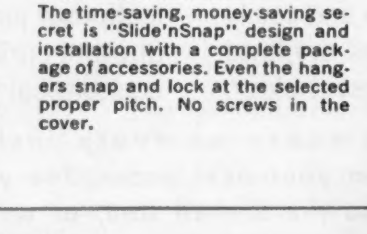
The new Fin Vector flush mounts, 1, 2 or 3 tiers high. It fits wall-to-wall installation like a glove. Or, Fin Vector can be efficiently employed in separate units as convectors and combined with Dunham-Bush Unit Heaters, Heating-Cooling Units and Convectors . . . for any combination of requirements.

Heating element size selection is a cinch as Dunham-Bush supplies a full range in copper and steel. Ingenious ball-bearing hanger provides "floating action", minimizing noise, at no extra cost.

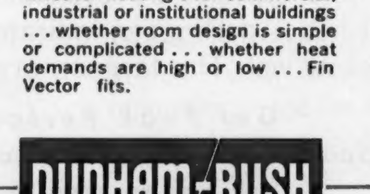
The new face and features of Fin Vector win quick approval of all concerned with heating . . .



Rounded Corners Complement Sleek Long-Low Design
Architecturally correct in all respects. Whatever the building construction or decoration, Fin Vector presents an attractive, continuous flowing appearance without overlap seams. Louvre design minimizes wall smudging.



Quick . . . Economical . . . Easy Lowest-cost Installation
The time-saving, money-saving secret is "Slide'nSnap" design and installation with a complete package of accessories. Even the hangers snap and lock at the selected proper pitch. No screws in the cover.



The Building Isn't Built That Poses a Problem for Fin Vector
Whether heating's for commercial, industrial or institutional buildings . . . whether room design is simple or complicated . . . whether heat demands are high or low . . . Fin Vector fits.

DUNHAM-BUSH

Send for complete specification details of Dunham-Bush "FC", "TFC" and "SC" Fin Vector Radiation. Request data sheets No. 1258C, No. 1271, No. 1272, No. 1273, and No. 1274.

DUNHAM-BUSH, INC.
WEST HARTFORD 10, CONNECTICUT, U. S. A.
SALES OFFICES LOCATED IN PRINCIPAL CITIES

(*Continued from page* 102)

motor or for direct-mounted applications utilizing the motor throughbolts. Included with both the permanent-split capacitor and shaded-pole motor are $2\frac{1}{4}$-in. diam mounting rings, which are assembled on the motor and mounting adapter. Extra ring adapters are available for special $2\frac{1}{2}$-in. diam resilient base mounting requirements.

Permanent-split capacitor motors are available in six-pole models from 1/15 through 1/4 hp, 115 or 230 volt; shaded-pole units are offered in four-pole models, 1/8 through 1/4 hp, 115 or 230 volt.

General Electric Company, One River Rd., Schenectady 5, N. Y.

URETHANE INSULATION

In addition to standard rigid urethane foam sheets, this manufacturer has introduced a flame-retardant coated version designed to minimize fire hazards during construction. Insulation is available in panels of various thicknesses up to 11 in. Blocks of foam suitable for use as pipe insulation and lagging also are offered.

Properties cited include the following: insulation can be applied directly with hot asphalt and many other adhesives; requires no special handling, cutting or shaping tools; is impervious to deterioration from rot and fungi; is dimensionally stable with low shrinkage and expansion factors; does not absorb water and maintains consistently low vapor permeability; and it displays a high strength-to-weight relationship.

Maximum recommended temperature for continuous use is 190 F. Compressive strength is 35 psi at 9%, flexural strength is 45 psi and tensile strength is 40 psi. K factor of the material is 0.15 at 70 F and 0.13 at 40 F.

Allied Chemical Corporation, Barrett Div, 40 Rector St., New York 6, N. Y.

FAN COIL UNITS

Five basic fan coil units for room air conditioning are designed for commercial, industrial or residential apartment buildings and provide year-round air conditioning for multi-room applications. Capacities of the new remote units range from 200 to 700 cfm. Each may be selected for either of four types of installation: two floor and two ceiling mounts.

One of the floor model choices is a cabinet type; the other is a recessed type designed for permanent hidden installation. Choice of cabinet or recessed model is offered also with the two ceiling units.

Chrysler Corporation, Airtemp Div, P. O. Box 1037, Dayton 1, Ohio.

P-TRAP

Fabricated from a single piece of seamless copper tube, this suction-line P-trap is cited as replacing cum-

bersome assemblies of individual fittings and promoting oil migration in refrigeration systems.

Oil is drained from horizontal runs approaching the risers, then migrates through the riser to the compressor in one of three different forms: as a rippling oil film, mist or transparent colloidal dispersion in the vaporized refrigerant. The method of oil migration depends on the vapor velocity in the suction line.

Three sizes of P-trap are offered: ⅞, 1⅛ and 1⅜ in. OD. Unit is ready to be soldered into the line.

Mueller Brass Company, Port Huron, Michigan.

CONTROL, INDICATOR

Designed for use with submersible and vertical-type sewage and sump pumps and with septic tank effluent pumps, the Vis-A-Trol automatic control system is combined with a depth measuring device which shows the depth of liquid in the pit. The pumping rate can be compared with the in-flow at all times without looking into the wet pit.

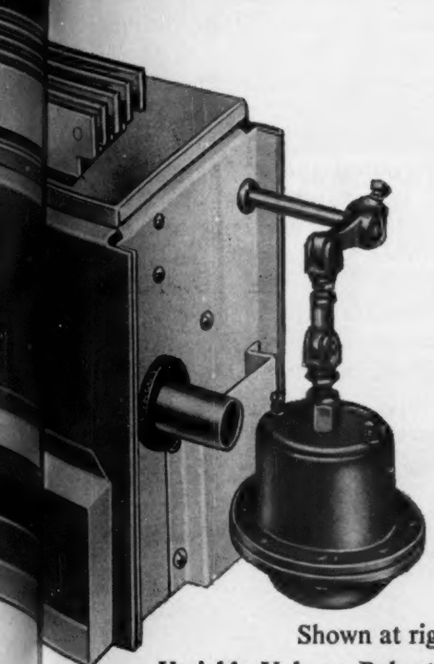
Installation can be inside buildings as far as 300 ft from the pumping pit or, where conditions dictate, can be in weather-proof enclosures on the outside. There are no electrical control connections between the wet pit and the control panel. Panel and instrument are a factory-wired unit including starters, alternator and high water alarms.

Weil Pump Company, 1530 N. Fremont St., Chicago 22, Ill.

Variable Volume Reheat unit by TITUS®

**developed in conjunction with Minoru Yamasaki... Smith, Hinchman & Grylls, Associated Architects and Engineers*

TESTED AND PROVED in a 2-story mock-up of the new Michigan Consolidated Gas Company Office Building in Detroit



PATENT PENDING

Shown at right is actual photo of new Titus Variable Volume Reheat units installed in mock-up of Michigan Consolidated Gas Company Office Building. The units were installed under the floor with a 3-inch pre-cast concrete sill containing Titus extruded aluminum Linear Grilles as outlets. *The new Titus VVR units fully met all requirements of the variable volume reheat system.* They proved capable of maintaining room temperature within 1 F—with varying heating and cooling loads.

MAIL COUPON FOR COMPLETE INFORMATION

TITUS MFG. CORP., WATERLOO, IOWA

Please rush new CATALOG giving complete details on the new Titus Variable Volume Reheat unit.

NAME____________________

COMPANY____________________

ADDRESS____________________

CITY__________ ZONE_____ STATE__________

Applications

AIR DISTRIBUTION IN A BOWLING ALLEY

Used in conjunction with 14 individual Arkla gas air conditioning units, the air distribution system for the 32-lane Pioneer Bowling Alley in Los Angeles circulates conditioned air through the alley approach area, where cool air is needed most. In addition, air returns are positioned behind spectator seating, causing fresh air to move from registers over the approach area through spectator areas, drawing smoke and other pollutants away from the bowling lanes.

CONDITIONING AN UNDERGROUND MISSILE LAUNCHING COMPLEX

In the Titan facilities at Vandenberg Air Force Base, Calif., the 97-ft missile is kept in operational readiness in a large circular hole in the ground, designated a missile silo. Closed over the opening are two hinged concrete doors several feet thick. Sunk in the ground on each side of this area are two smaller silos, linked to each other and to the third by a ten-ft diam underground tunnel.

In this facility, the chilled water for cooling and hot water for heating are supplied from a powerhouse which is also entirely underground. Supplying chilled water are two Trane 300-hp hermetic CenTraVacs. Water is circulated continuously through the entire system to assure that it will be available to any facility without a time lag. To accomplish this, bypass systems are used, with pressure differential valves that will open when the demand drops and modulate down to 10% of capacity as the demand increases.

Used as a heat sink is a spray pond consisting of two concrete basins, each capable of rejecting the heat for the entire system, including cooling water to diesel generators. This capacity is obtained by spraying approximately 2000 gpm of water through 50 nozzles at 15 and 20-psi pressure. Two horizontal Scotch marine-type low pressure steam boilers are used for production of required heat. Each has a capacity of 125 hp and is capable of handling the full load. Low pressure steam is utilized as the heating medium in the powerhouse and hot water, circulated through a steam-to-hot-water heat exchanger, is used throughout the other facilities.

Each area is served individually by one or more Trane Climate Changer central station air handling units, with chilled and hot water coils and water sprays as required to control the dry bulb temperatures and relative humidities. During normal operation, the units maintain a 75 F dry bulb temperature and 40 to 50% relative humidity. During peak loads the temperature is permitted to rise to 95 F. Air distribution in the missile silo is controlled by a number

of duct rings around the inside circumference at levels that will provide the best possible conditions for both a normal and purging air supply.

30-TON COOLING FOR CALCULATOR MACHINES

To control temperature and humidity in the IBM Department of Trailmobile, Inc., in Cincinnati, two

Dunham-Bush, 15-ton packaged air conditioning units were selected. Compressors were installed in a remote location to reduce the noise level, and the two 15-ton LRCU large remote condensing units were mounted on the roof of the office building. Shown in background and right foreground, under ductwork, is the equipment.

NUCLEAR SUBMARINES TO USE AIR-CLEANING PRECIPITATORS

To provide clean air for personnel to breathe during extended periods underwater, the U. S. Navy is having air-cleaning precipitators installed in its nuclear submarines. Capable of cleaning nearly 1,000,000 cfm of air, the units are custom designed to fit the submarines and have many different configurations. All are of the two-stage type, having a charging section of 17 kv and a collecting section of 7 to 17 kv. To meet minimum clearing requirements, the units will be negatively polarized. Depending on available space, new high velocity precipitators are capable of high removal efficiency in the range of 90 to 95% of the particles from air moving at gas velocities ranging from 10 to 35 fps.

HIGH VOLTAGE UNIT SAVES COSTS

Growing in favor is use of refrigeration machines which operate on high input voltage, resulting in savings from elimination of the transformer capacity for the refrigeration compressors and reduced wiring costs. In a cited installation in the Federal Office Building, Albuquerque, N. M., two 250-ton American-Standard Tonrac centrifugal refrigeration machines were selected for the air conditioning system. These hermetic units operate from a 4160-volt electrical power input.